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TO

DEPARTMENT OF ENERGY
WAVE ENERGY STEERING COMMITTEE

SUPPLEMENTARY REPORT
ON THE FEASIBILITY STUDY
OF

SUBMERGED CYLINDER WAVE ENERGY DEVICE

Submitted by
Sir Robert McAlpine & Sons Ltd.
Dr. D. V. Evans, University of Bristol

MARCH 1980

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NOTATION

a	cylinder radius
c	piston amplitude (= half piston stroke); also submergence (=h-a)
c_1	cylinder position
d	water depth
g	gravitational acceleration
h	mean depth to centre of mean cylinder position
h_1	head lost in pumping main
k	wave number (= $2\pi/L$)
ℓ	length of wave energy device parallel to wave crests
q	ratio of maximum capture width of each of a number of equally spaced devices to that of an isolated device
r, t	wave reflection and transmission coefficients
s	wave steepness (= H/L)
u, v	orthogonal velocity components
w	unit weight of fluid
x, y	directions along orthogonal fore and aft rodes, respectively
A	cross-sectional area of flow of pumping main
B	damping coefficient of device
D	cylinder diameter
Do	internal diameter of pumping main
F_H, F_V	horizontal and vertical forces
H	wave height
Hp	pressure head in pumping main
L	wavelength
M	length of pumping main
P	power absorbed per unit crest length
Q	discharge through pumping main
T	wave period
Tf	tonne force (= 1000 kgf)
V	average flow velocity in pumping main
β	wave angle to normal to axis of wave energy devices
ϕ	phase angle
λ	friction coefficient for flow in pumping main
ρ	fluid density
ω	angular frequency (= $2\pi/T$)
χ_s	complex amplitude of exciting force on a fixed body
η	efficiency of absorption of wave energy

EXECUTIVE SUMMARY

This report has been compiled as the result of a four-month supplementary contract to our first broad look at submerged cylinder wave energy devices.

In this further period we have discussed widely within the wave energy community the findings of our main Report dated October 1979. We have also considered topics that were either not within our earlier brief or were only peripheral to its principal purpose of identifying how the device could be engineered using proven technology, and the likely range of unit costs that this could involve.

The present Report therefore covers a wide spectrum of topics. These are mostly pitched in the form of 'Appendix' notes to subjects studied and presented in our main Report. They serve to clarify points of detail, and have been pursued in sufficient depth to show whether they need be studied further in the next phase of this enquiry.

We have also continued our studies of the main subjects underlying the behaviour and performance of the device, especially its dynamic behaviour in response to wave motion and how energy is best transmitted to and through the seabed power takeoff units.

This supplementary period has therefore allowed us to narrow our options. Although we believe it is still premature to expect the preferred system of mooring and power takeoff to be selected with certainty, our earlier recommendations are upheld by the further findings now presented. This system will therefore form the basis for the optimisation study of the device that now logically follows, but we will continue to seek improvements both to the overall arrangement of the device and to its component parts in the light of all further information that becomes available to us.

We conclude that the submerged cylinder device is a technically sound and efficient way of capturing wave energy. On the basis of present knowledge we have reason to believe that, from the thorough optimisation study that constitutes the next phase of our work, the device may also turn out to be an economic proposition. In this case it should be advanced through a full engineering design phase to the prototype construction of perhaps five units at full scale in say 1984/5.

SECTION 1

INTRODUCTION

Our first overall appraisal of the submerged (or 'Bristol') cylinder device was submitted to WESC in October 1979 after an investigation lasting seven months. In that period we sought to make estimates of the quantity, the value and the cost of electrical energy delivered per annum to the network at Perth by a nominal 2GW installation of cylinders off South Uist.

During that formative period we were more concerned with identifying and comparing possible solutions for the principal elements of the device based on proven components and technology than for the many innovations in design and assembly which the device, because of its repetitive modular form, readily allows.

The present Report is written following wide circulation and discussion of the content of our earlier Report within the wave energy community. It aims to fulfil two main functions, firstly to assemble and appraise the many reactions we have had to our earlier ideas and, secondly, to present the early results of new work started in support of the next major round of studies of the device during which its design and performance must be advanced to an optimised engineering concept.

Each element of the device, from the wave climate incident upon it through its energy conversion and structural forms to its maintenance problems are dealt with in further detail in the present Report. The continuing flow of new ideas as to how best the system should be designed and operated has encouraged us to consider substantial changes to component parts of the device, though its overall form remains essentially unchanged from that proposed earlier.

Detailed attention has been given to the moorings, anchors, hydraulic power transmission systems and to several aspects of marine fouling that appear likely to influence some aspect of the performance and hence the design of the device.

The energy capture efficiency of the device when tuned to match more nearly the annual S. Uist wave climate has been studied in detail, including reference to the preferred dimensions for the cylinders, their submergence, spacing and local water depth. The results will help to guide the forthcoming tank tests when optimum performance in mixed seas will be identified. It will then be timely to review costs as part of the optimisation phase that now logically follows completion of the broad appraisal in the present Report of all major aspects of the device known to us at present.

SECTION 2

ASPECTS OF CYLINDER BEHAVIOUR AND MOTION

2.1 Response of Device in Waves

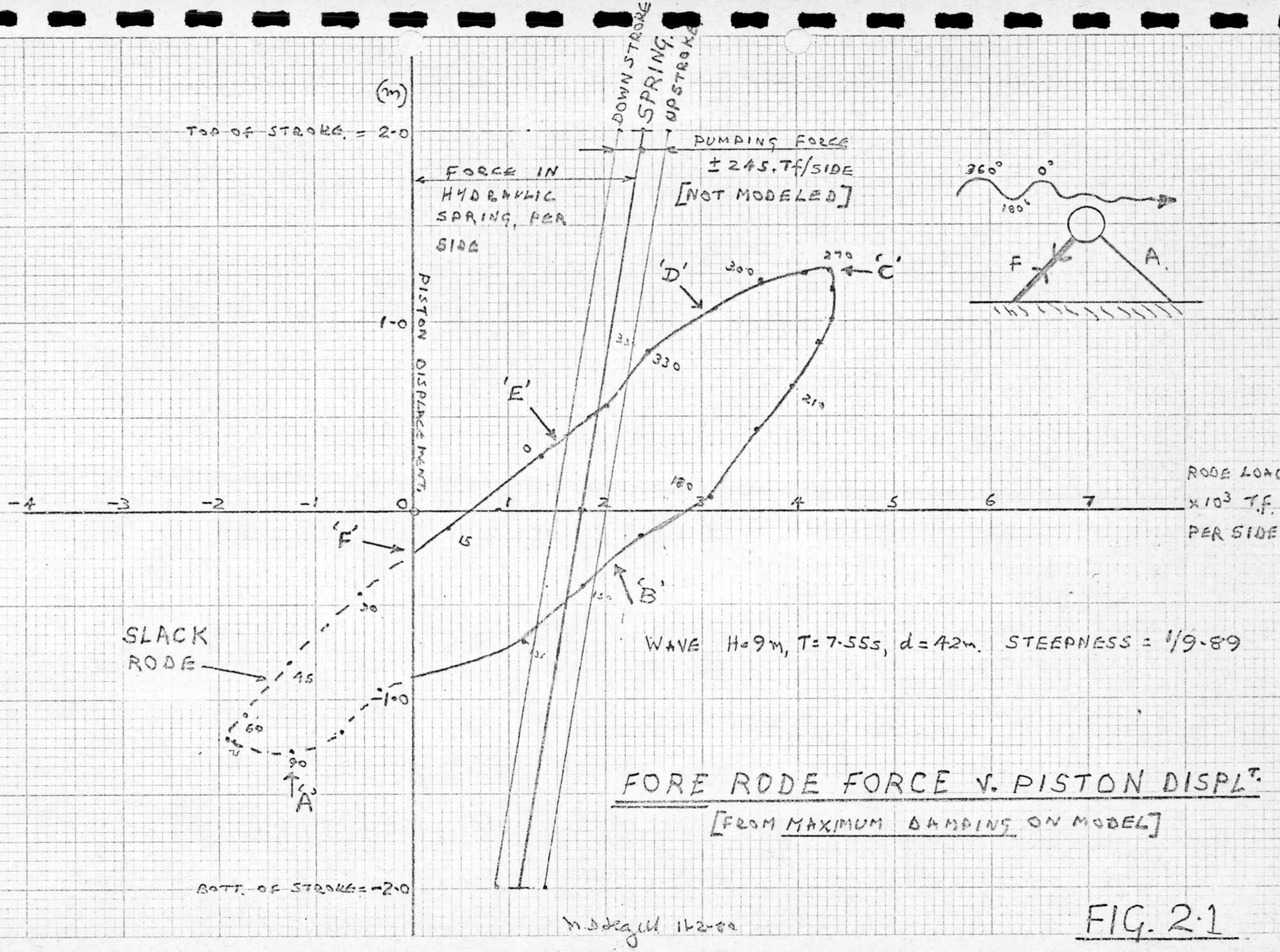
Our Report to WESC dated October 1979 shows, in Figure 2.1B, the surface wave profile, model cylinder orbit and rode motions in a 1:120 scale model wave of height 75mm, period 1.45 Hz and steepness 1/9.87.

By using Stokes 2nd Order Waves and Morison's Equation, these motions have been used to determine the full scale motions and forces for the two conditions of damping that gave the diagrams referred to above, but which were not quantified.

This study produced diagrams of rode force v. piston displacement (area = work done) in a survival sea. For maximum damping the fore rode 'footprint' of power output is as shown in Fig. 2.1, and the aft rode footprint is shown in Fig. 2.2 of the present Report.

The following events that occur during a wave cycle with a properly loaded cylinder/pump/spring combination (see e.g., Fig. 6.4 of Oct. '79 Report) are described using the lettered reference points shown on Fig. 2.1.

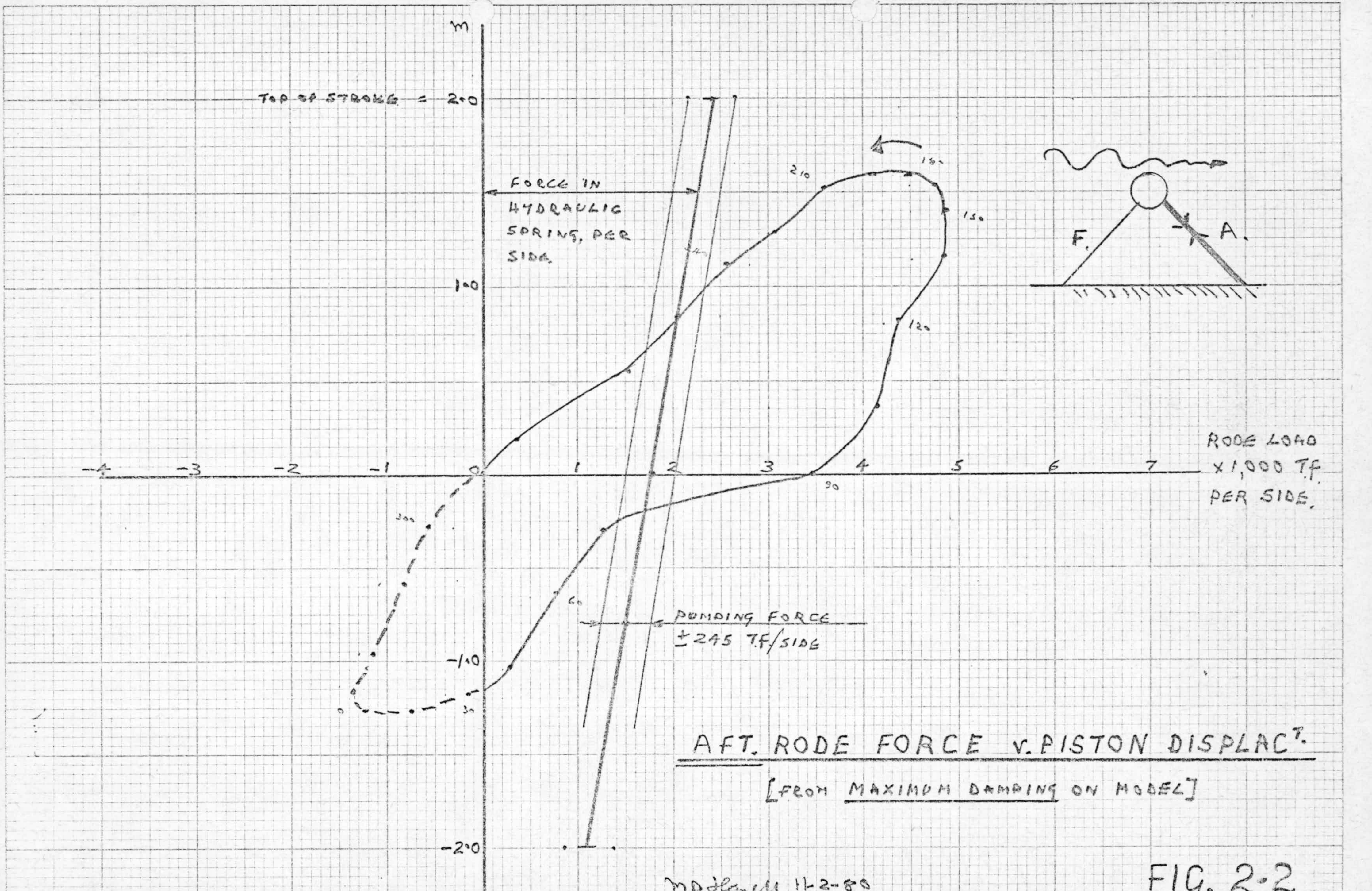
1. During the up-stroke (work done by cylinder). Point A (bottom of modelled stroke). Piston cannot rise until the rode force is equal to the combined spring and pump forces.
   Point B. Rode force is now greater than the combined spring and pump forces. Piston rises, pumps, and compresses spring.
   Point C. Upward motion of piston stops. Pumping and spring compression stop.
   (Note that the elevation potential of a buoyant cylinder is lost during this phase of the motion).



FORE ROPE FORCE V. PISTON DISPL^{T}.

[FROM MAXIMUM DAMPING ON MODEL]

FIG. 2·1



AFT. RODE FORCE v. PISTON DISPLAC[T].

[FROM MAXIMUM DAMPING ON MODEL]

M.D. Haagell 11-2-80

FIG. 2·2

2. During the down-stroke (work done by spring)

   Point D. Combined spring and pump force is less than the rode force because the cylinder holds the piston up. Hence there is no piston motion and no pumping.

   Point E. Rode force is less than the difference between the piston and pump forces. Spring drives the pump and helps to pull the cylinder down (depending on buoyancy - See Sect.8.3) in the downward phase of wave force application.

   Point E. Rode force is zero. Spring continues to drive the pumps until bottom of stroke reached at A. Cycle of events complete.

This simplified statement of the component parts of an orbit covers its principal features but ignores some important details. For example, between points B and C there is apparently an excess of rode force over that which is absorbed between pumping, accelerating the cylinder through the water and causing energy losses between wave and cylinder when the motion of the latter is damped. It is suspected that the experimental data presented in Figure 2.1B of our October 1979 Report, and used as a basis for Figs. 2.1 and 2.2, are misleading for the present purpose in that they were recorded in conditions that did not properly simulate the pumping mode as currently envisaged. Experiments that include this mode, and that also have adjustable damping to simulate various possible spring characteristics, are to be carried out in mid-1980 (see Appendix B for details). It will then be possible to quantify the analysis introduced in Figs. 2.1 and 2.2, the procedure for which is explained in Section 2.2 below. In this case, because fully relevant experimental data were not available, the cylinder motions were calculated from first principles, taking into account the wave/cylinder/mooring/pump/spring/fender (cushion) interations described in more detail below.

2.2 Analysis of System Behaviour and Motions

The theoretical motions and forces produced by a prototype cylinder in survival waves, taking account of the various interactions between the wave/cylinder/moorings/pumps/springs and cushions were analysed as follows :

The fore and aft rodes are inclined at 90° to each other, and are taken to constitute a pair of orthogonal axes x (fore rode) and y (aft rode). The following symbols are also used :

$x, \dot{x}, \ddot{x}$ = wave position, velocity, acceleration on x-axis

$y, \dot{y}, \ddot{y}$ = wave position, velocity, acceleration on y-axis

$c, \dot{c}, \ddot{c}$ = cylinder position, velocity, acceleration (2c= piston stroke

$c_1, \dot{c}_1, \ddot{c}_1$ = calculated cylinder position, velocity, acceleration

ϕ = phase angle (ϕ = 0° at wave crest)

d = water depth

g = 9.81 m/s^2

Tf = tonne force

2.2a Basic Conditions

1. Use the same steep prototype wave as in Section 2.1, H = 9m, T = 7.55 secs, d = 42m, depth to centre of cylinder in still water = 3m (submergence) plus 6m (cylinder radius), L = 88.7m, steepness = 1/9.87), for direct comparison.

   Use Stokes 2nd Order Eqns to find the horizontal and vertical velocities (u,v) and accelerations ($\dot{u}$, $\dot{v}$) for 15° increments of phase angle ϕ through a wave cycle.

   Resolve u,v and $\dot{u}$, $\dot{v}$ along the fore rode to find $\dot{x}$, $\ddot{x}$.

2. Morison Drag Force = $41|\dot{x} - \dot{c}|^2$ Tf.

3. Morison Inertia Force = $1182(\ddot{x} - \ddot{c})$ Tf.

4. Cylinder weight in air = 3284 Tf.

5. Cylinder buoyancy = 1778 Tf/side, assumed constant in the present calculation (ref. Section 8.3).

6. Force accelerating cylinder = 3284 $\ddot{c}/g$ = 335 $\ddot{c}$ Tf.
7. Spring force at mid stroke = buoyancy = 1778 Tf/side.
8. Spring stiffness = 430 Tf/m/side.
9. Spring + cushion characteristics as shown in Fig. 2.3

   Piston stroke = 4m (but see Section 5.6).

11. Bus main contents acceleration plus pumping force = $\pm$260 Tf/side.

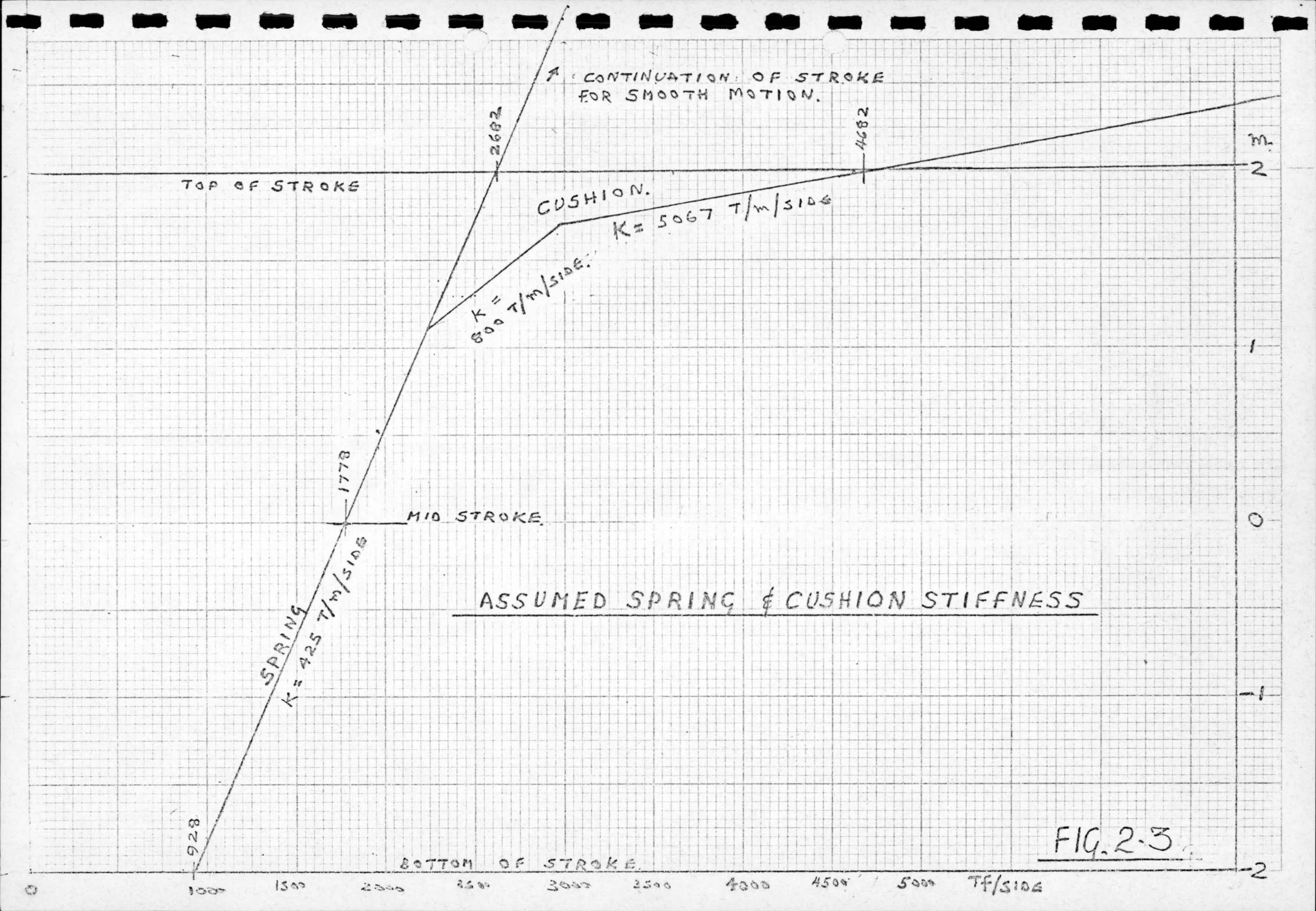
2.2b Starting Conditions

Assume the fore pistons are at the bottom of their stroke and the fore rodes are under a slight tension. The piston will not start to rise until the rode tension just exceeds the combined spring and pump force (= 928 + 260 = 1188 Tf). This is when the combined buoyancy and Morison forces = 1188 Tf, or when the Morison forces are just less than 590 Tf "pushing down" (relieving the buoyancy tension in) the rode. This occurs at ϕ = 123° on a fixed cylinder.

The calculation therefore starts at ϕ = 120°, when c = −2.0m, $\dot{c} = \ddot{c} = 0$ 15° intervals are taken.

2.2c Procedure for Step-by-Step Calculation

1. Guess a value of $\ddot{c}$ at ϕ = 135°.
2. Calculate c and $\dot{c}$ at 135°.
3. Determine spring/cushion force from c.
4. Determine Morison Drag Force (= $41|\dot{x}-\dot{c}|^2$).
5. Determine Morison Inertia Force (= 1182 ($\ddot{x}-\ddot{c}$)).
6. Determine upward force at top of rode (= 4+5 above + buoyancy).
7. Determine downward force at bottom of rode (= 3 above + pumping force).
8. Determine difference of axial rode forces. If difference is positive (acting up fore rode), this is the absence of downward restraint that allows the cylinder to accelerate upwards due to its buoyancy.



CONTINUATION OF STROKE FOR SMOOTH MOTION.
2682
4682
m.
2
TOP OF STROKE
CUSHION.
K= 5067 T/m/SIDE
K = 800 T/m/SIDE
1
1778
MID STROKE
0
ASSUMED SPRING & CUSHION STIFFNESS
SPRING
K= 425 T/m/SIDE
-1
928
BOTTOM OF STROKE
-2
0
1000
1500
2000
2500
3000
3500
4000
4500
5000
Tf/SIDE

FIG. 2-3

9. Determine $\ddot{c}_1$ (= cylinder acceleration) = Difference/ $335m/s^2$
10. If $\ddot{c}_1 \neq \ddot{c}$, revise $\ddot{c}$ and repeat steps 2-9 above inclusive. Repeat as necessary until fair agreement reached.
11. Move to ϕ = 150° and repeat calculation steps 1-10 above for instantaneous values.
12. Continue through increments of phase angle and through consecutive identical waves until results are repetitive. (In the present calculations the upper stroke limit was 'over run' to avoid discontinuity).

The calculated forces and displacements through three consecutive waves are shown in Figs. 2.4 - 2.6. The 'footprint' of pump and spring force v. piston displacement is shown in Fig. 2.7, from which the acceleration force is omitted for clarity.

If the assumption is made that the motions and forces in the aft rode are identical with those in the fore rode, but 90° earlier in phase angle, a theoretical cylinder orbit can be plotted. This is shown in Fig. 2.8. (The validity of this 'similar but phase shifted' assumption will be checked when further experiments are carried out : the calculation procedure will be adjusted if necessary.

The orbits shown in Fig. 2.8 display the expected two 'corners' or arcs of reduced radius. It also shows a considerable shift in the phase angle at the top of the orbit, from 135° for the 75mm high <u>model</u> wave (9m prototype) with minimum damping, or 225° for the 75mm model wave with maximum damping (Fig. 2.1 of Oct. '79 report), to 315°.

2.2d Discussion of Method of Analysis

The method described in Section 2.2c can be used for any wave, any pump force, and any reasonable spring/cushion characteristic. It can be used to describe the behaviour of the system throughout two or more different consecutive smooth waves by initially finding the motion that satisfies the basic relations in regular waves of the <u>first</u> type.

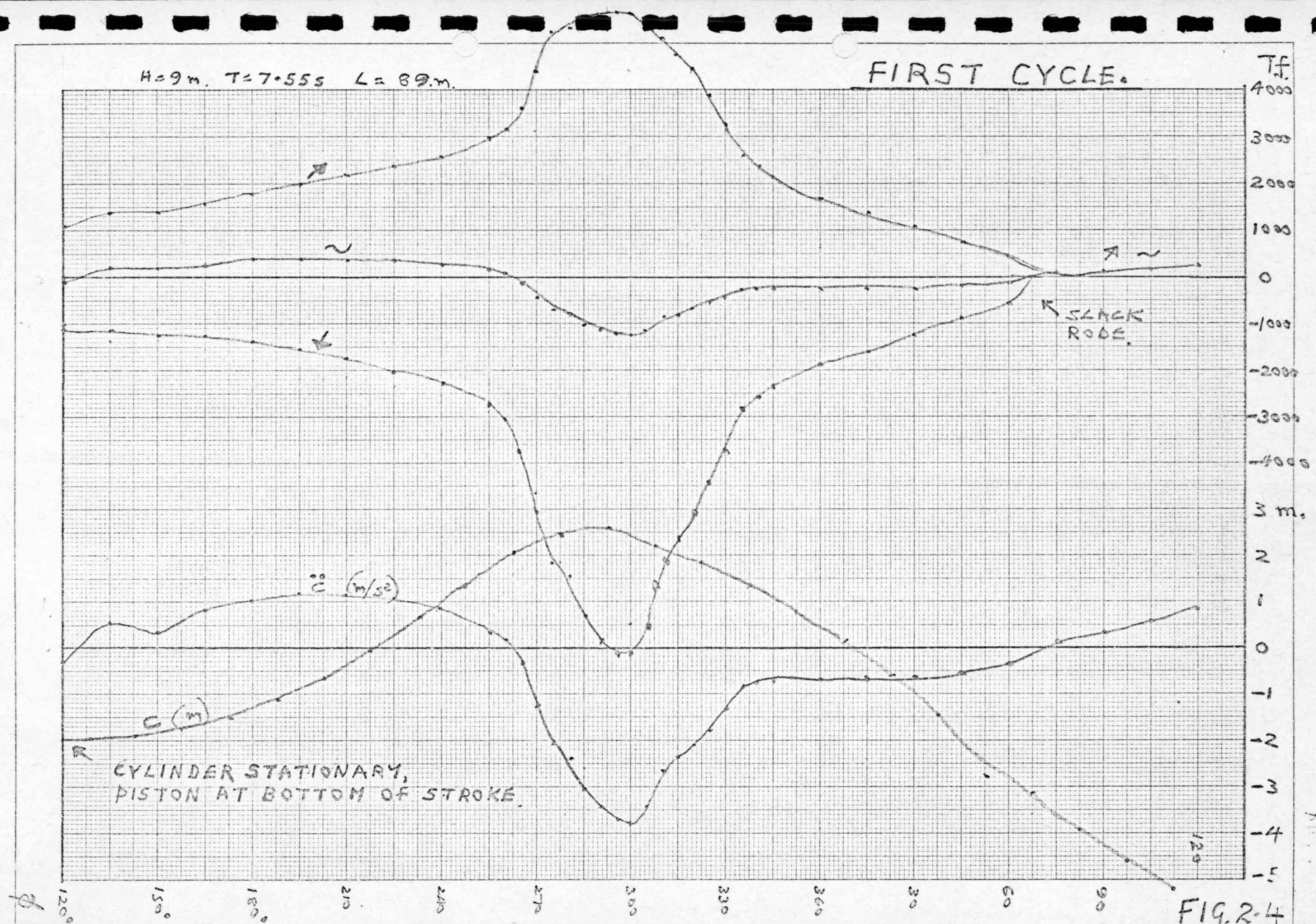

H=9m. T=7·55s L=89.m.
FIRST CYCLE.
Tf.
4000
3000
2000
1000
0
-1000
-2000
-3000
-4000
3 m.
2
1
0
-1
-2
-3
-4
-5
SLACK RODE.
ü (m/s²)
c (m)
CYLINDER STATIONARY,
PISTON AT BOTTOM OF STROKE.
φ
120°
150°
180°
210
240
270
300
330
360
30
60
90
120

FIG. 2·4

SECOND CYCLE.

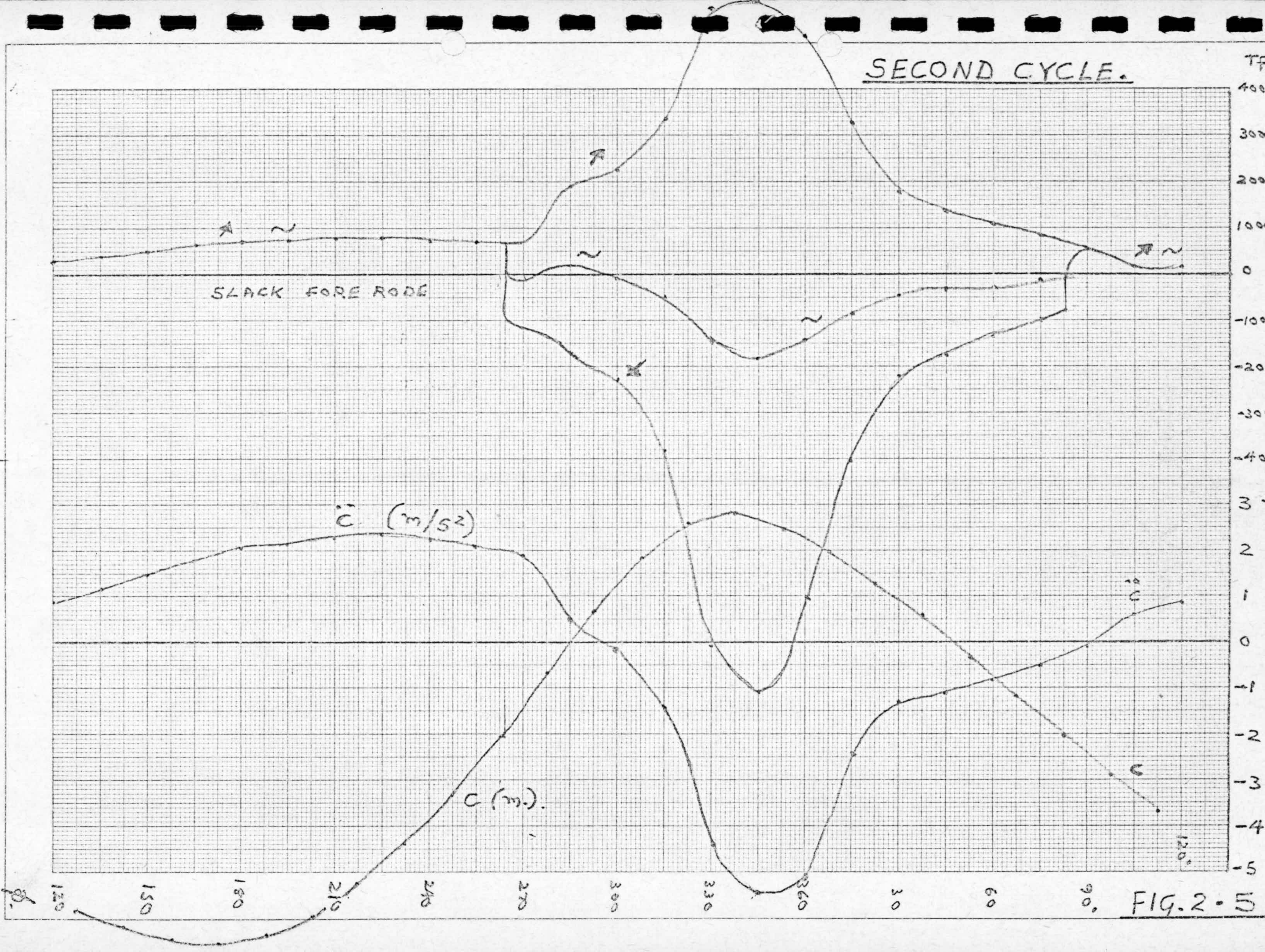


FIG. 2·5

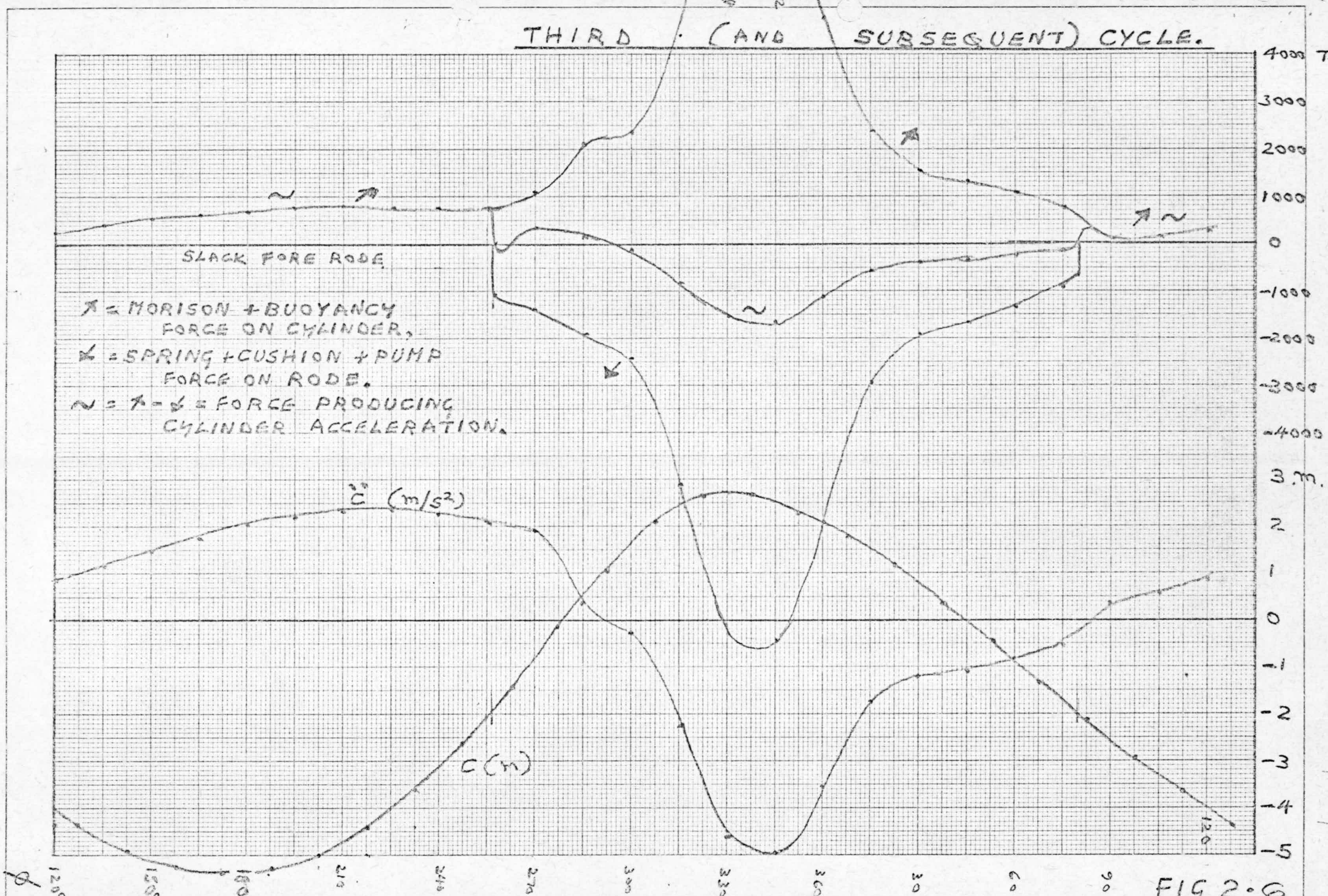
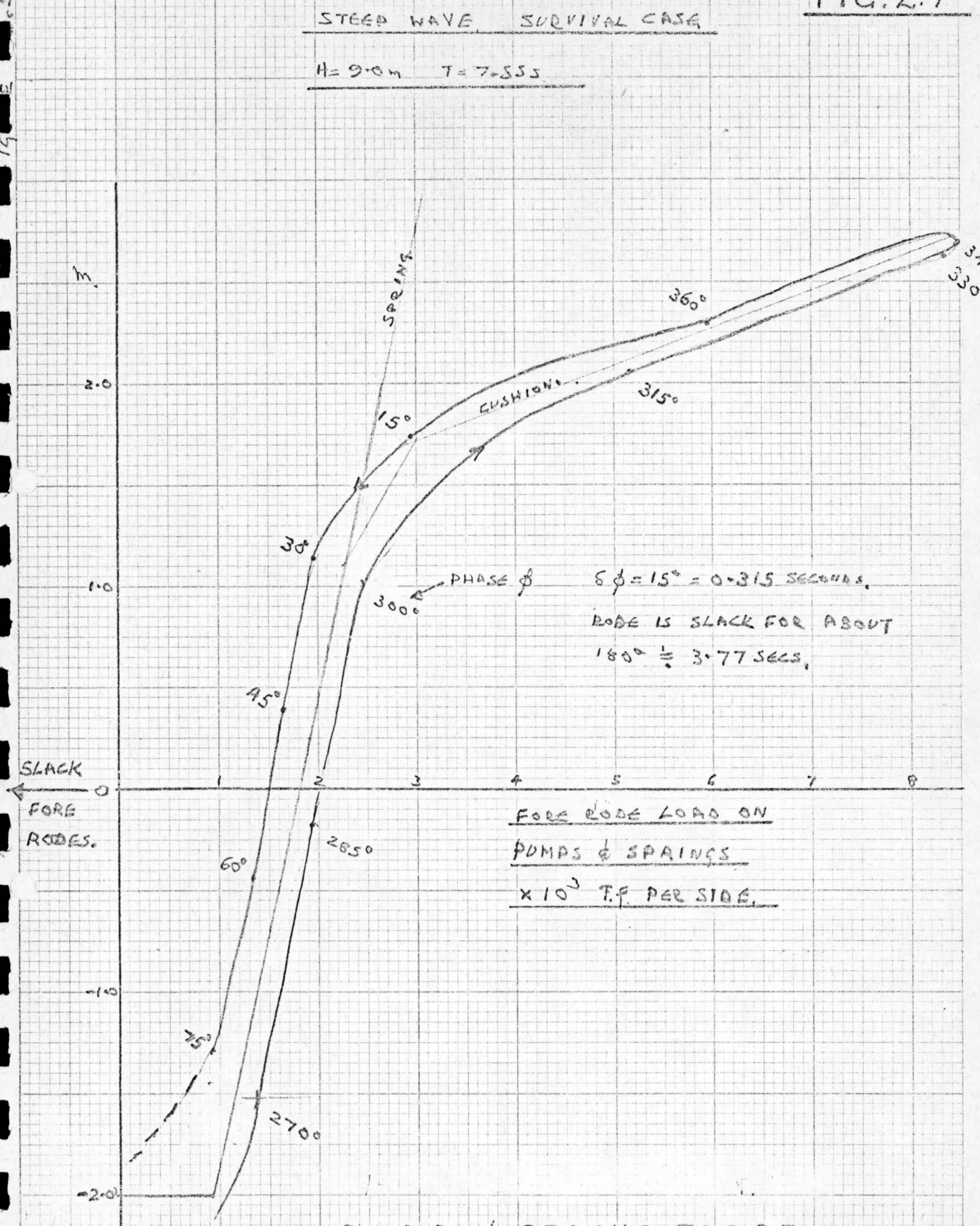


FIG 2·6

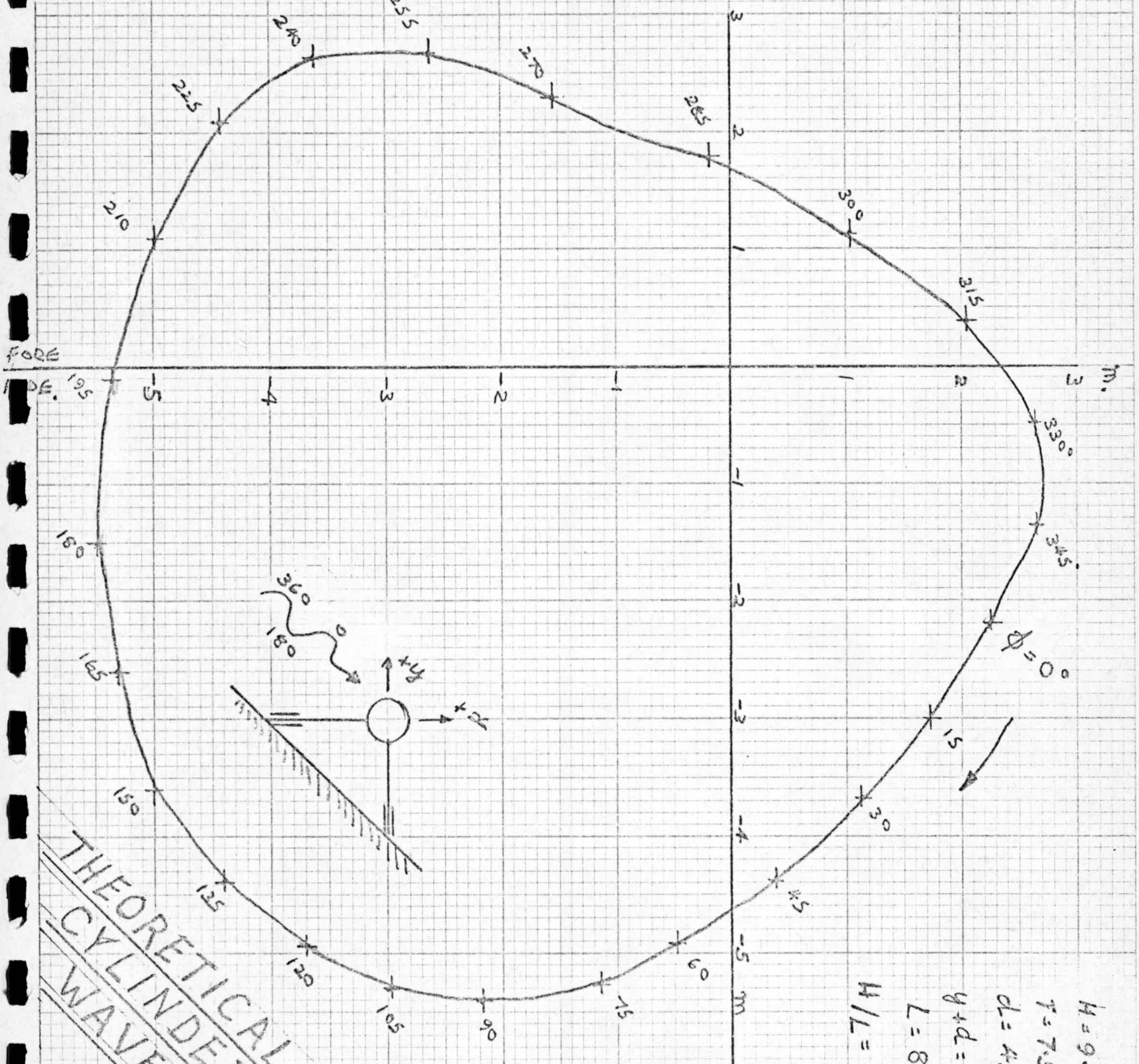
M.D. Hogell. 31-3-80

FIG. 2.7



PUMP & SPRING FORCE V.
PISTON DISPLACEMENT.

ϕ	FORE RODE.	ϕ	AFT RODE.
90°	SLACK.	0°	SLACK.
265	TAUT. PUMPS ON UPSTROKE	175	TAUT. PUMPS ON UPSTROKE
315	OVER RUN.	225	OVER RUN.
340	TOP OF STROKE ! PUMPING STARTS ON DOWN STROKE.	255	TOP OF STROKE. PUMPING STARTS ON DOWN STROKE.
90°	SLACK.	0	SLACK.



THEORETICAL ORBIT OF CYLINDER IN STEEP WAVES. SURVIVAL CASE.

Scale 1/50 m

The method does not yet take account of the cylinder moving across streamlines. This can be done, step-by-step, with discrete adjustments to $\dot{x}$ and $\ddot{x}$. It is also not clear how the wave forces acting on a large cylinder in water motion, that in the absence of the cylinder decays exponentially with depth, will differ from those that are assumed here to act at its centre, nor how the presence and movement of the cylinder will modify those motions as far as the forces carried by the device are concerned. It is expected that the experiments on device performance shortly to be carried out will also throw light on these points.

The calculations reported in Figs. 2.3-2.8 were made on a pocket calculator. The work could be programmed for a large computer with an extensive storage ability. This programme could be refined to show the combinations of wave height and period that would produce the most violent decelerations at the top of the stroke, and how these depend on the length of stroke permitted (see Section 3.1).

2.2e Discussion of Results

The calculated motions and forces based on the procedures and data set out in Sections 2.2a-2.2c are considerably in excess of those reported in Section 2 or our October 1979 Report, which were based on experiments. This is attributed mainly to the different wave steepnesses used in the two methods. A flatter wave would reduce the motions and forces. It is important to resolve how the return probability of waves of any prescribed steepness vary with wavelength, hence what should be the design steepness/wavelength combinations for the cylinder device. Section 2.3 below sets out our observations to date on this urgent subject. By adopting for design a wave steepness s_1 higher than the correct value s_2 (at the same wave height), the wave particle acceleration is increased by $(s_2/s_1)^2$; the forces imposed on the cylinder device by these waves are similarly related (see also Section 3.1).

2.3 Hydrodynamic Information relevant to Wave Forces

Our studies have covered a number of areas that could not be considered in detail in the period prior to completion of our October 1979 Report. Several of these are presented in this Section. Others appear where most appropriate throughout this Report, for example wave directionality is in Section 8.3 on energy capture estimates, and crest lengths and backscatter in Appendix A.

2.3a Wave Steepness

In our earlier Reports (June and October 1979), we expressed concern about the wave steepness variously quoted for the design of offshore structures, and in the final paragraph of Section 2.2e above we point out the importance for accurate and economic design of having a more rational foundation for this parameter.

On the basis of the very limited information available to us on wave steepness we have provisionally concluded that, for the longer wavelengths to which structures like ships and the cylinder device are more sensitive, a steepness certainly not exceeding 1/10 should be used. The available evidence suggests that a considerably flatter wave would be more appropriate. Brief discussions have been held with TAG4 and staff of BP regarding wave height and length. Mr. S.M. Abdi, of BP, is of the opinion that 1/10 is probably too steep a requirement for the present purpose. BP have extensive information from the North Sea that gives a steepness of 1/16 to 1/20 in storm conditions. They also have confidential information from 30 miles offshore of Foula which confirms this for greater wavelengths. BP's North Sea practice would be to use a design steepness of 1 in 18, and certainly not the 1 in 7 value suggested in Lloyds report to WESC (Ref. 1).

Fig. 2.9 (Ref. 2) presents many of the proposed equations estimating wave steepness. The theoretical maximum line is quoted as having a steepness of 1/7 and is seen to approach the available records for waves less than 50m long. Also shown on Fig. 2.9 is a line representing the dependence of Morison Inertia Force, which is the principal wave force component on the

WAVE HEIGHTS FOR SHIP BENDING MOMENT CALCULATIONS

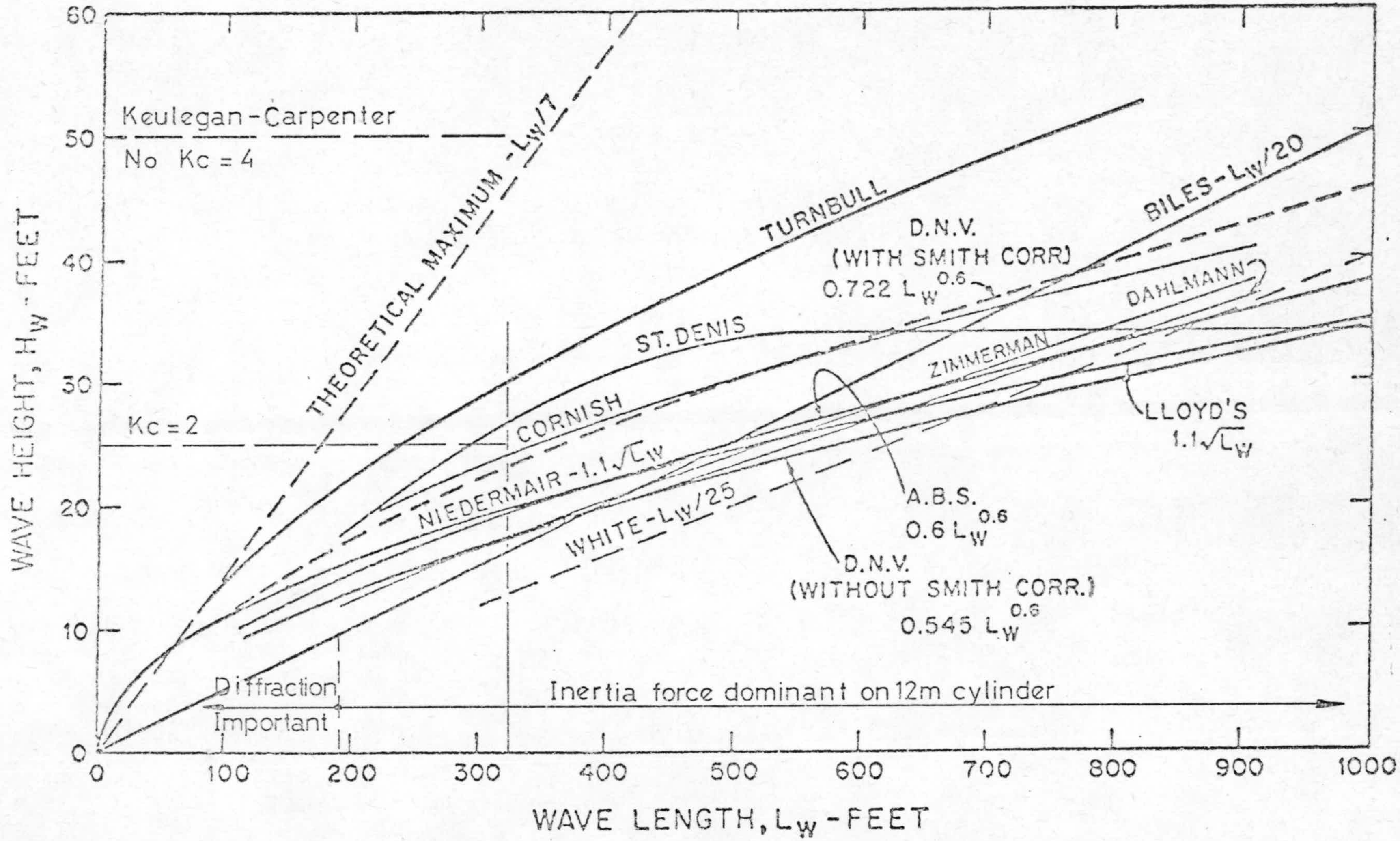


FIG 2.9

From ref. 2.

cylinder. It is clear that an important aspect of wave loading occurs at wavelengths less than the longest to be expected if the latter are less steep than may occur in shorter waves. The importance of the high wavelenth end of the steepness/wavelength relationship will then be of principal interest to the design of the stroke limiting device (fender or cushion) included to reduce the throw of the pistons in higher waves. A first attempt at predicting the relationship of throw to wave height was given in Fig. 2.2 of our October 1979 Report : a diagram like Fig. 2.9 herein would allow throw to be related to wavelength, hence via typical annual wave scatter diagrams to the probability of its occurrence.

The factor of wave steepness is therefore of vital importance to the design of wave energy devices, and the variations referred to above will have very significant effects on design and costs. As Fig. 2.9 herein shows, the Turnbull curve quotes a minimum length of under 100m for the 9m wave (steepness = 10.8) taken in Section 2.2a, whereas the Lloyds curve gives a steepness of about 9/225 = 1/25. By contrast Ref. 1 suggests a steepness of 1/7.

We question the value to this scientific debate of wave steepness of yachtmen's experience such as during the 1979 Fastnet Race, because the principal interest for the design of the submerged cylinder device is not the steepness that wave crests may attain (surface mounted devices may need this information) but the ratio of wave height to length which is, we suggest, impossible for an observer floating in asymmetric waves to estimate with sufficient accuracy.

2.3b Mean Second-order Vertical Force on a Submerged Cylinder

Ogilvie (Ref. 3) has shown that the second-order mean vertical force can be determined purely from a knowledge of the first-order velocity potential. In his Paper, this force is given by Eqn. 28a, b and it is sketched in his Figure 4 on page 459. The curve is not easy to interpret. It has been re-drawn here as Fig. 2.10, and it applies to both a general cylinder and our particular cylinder 50m long, 12m diameter, at different submergences.

Mean Second-Order Vertical Force On Fixed Submerged Cylinder.

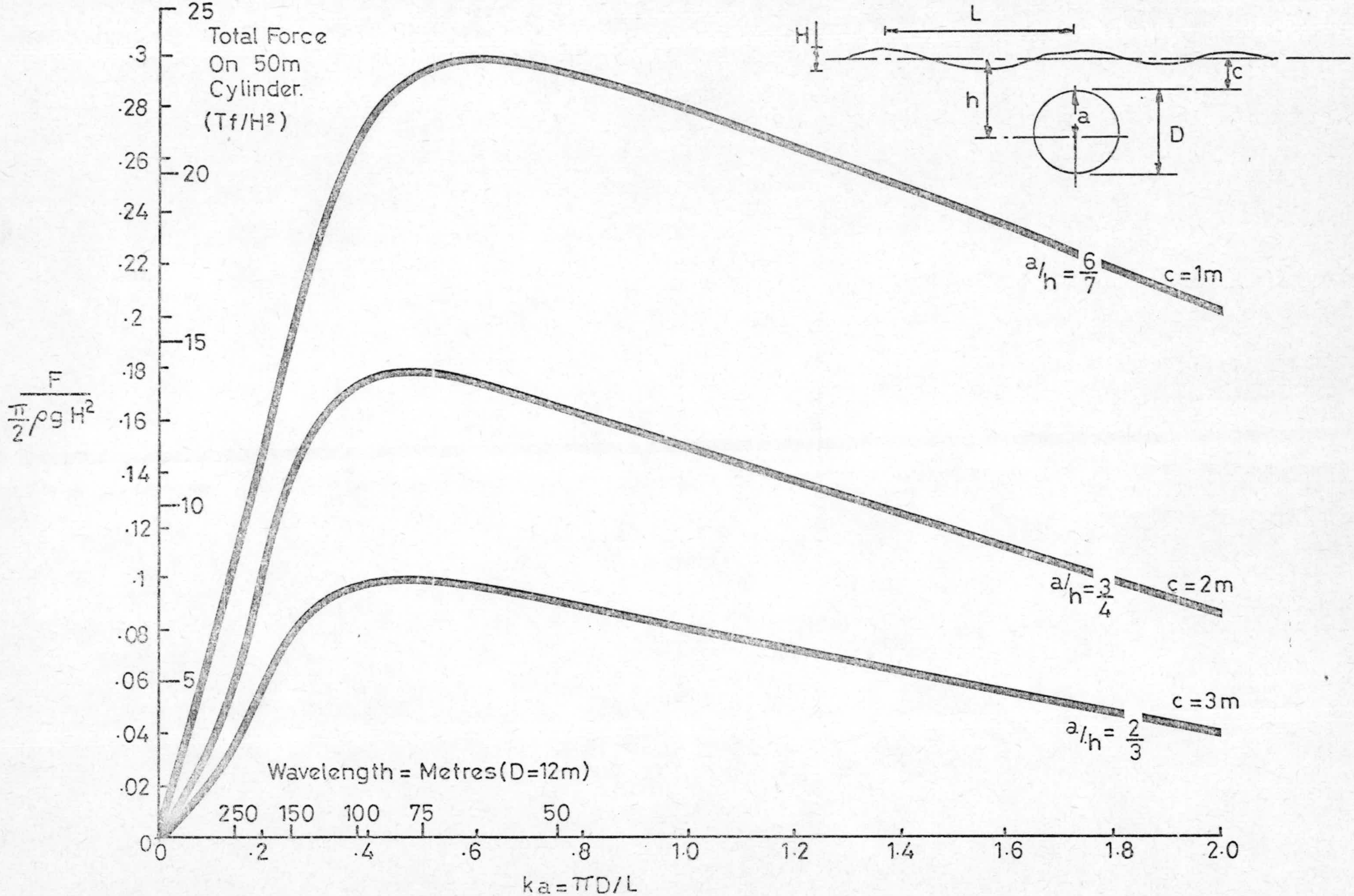


FIG 2.10

In his Paper, Ogilvie states that the information on which Fig. 2.10 is based requires that "the amplitude of the incident waves must be much less than the clearance between the top of the cylinder and the undisturbed free surface if the linearised theory is to have meaning". The significance of this statement by Ogilvie is referred to at the end of this sub-Section.

With reference to Fig. 2.10, clearances above the cylinder to still water level of 1, 2 and 3m are equivalent to values of $a/h = 6/7$, $3/4$ and $2/3$ respectively, where h is the mean depth to the centre of the fixed cylinder of radius a. Thus $2kh = 7ka/3$, $8ka/3$ and $3ka$ respectively (where k = wave number $2\pi/L$, L = wavelength : Ogilvie uses ν for k). For each clearance the value of the non-dimensional vertical force F_v can be estimated from Fig. 2.10, as follows.

$$F_v = \overline{Y_1^{(2)}(t)}^{\,t} / \tfrac{1}{2}\pi\rho g H^2 \text{ where } H = \text{wave height}$$

For a cylinder of diameter D = 12m, length 50m,

$$Y_1^{(2)} = \frac{\pi}{2}\rho g H^2 F_v \text{ per metre length}$$
$$= 1.57 \times 10^3 \times 10 \times H^2 F_v \quad \text{N/m}$$

The total vertical force on a 50m cylinder is

$$TF_v = 78.5\ H^2 F_v \text{ tonnes}$$

Table 2.1 sets out the relationships between ka and F_v (in tonnes) for the three values of a/h given above.

Table 2.1

ka	a/h	2kh	F_v	a/h	2kh	F_v	a/h	2kh	F_v
0.1	6/7	0.23	0.08	3/4	0.27	0.035	2/3	0.3	0.02
0.2	"	0.47	0.15	"	0.53	0.1	"	0.6	0.06
0.5	"	1.17	0.30	"	1.33	0.18	"	1.5	0.10
1.0	"	2.33	0.28	"	2.7	0.15	"	3.0	0.08
2.0	"	4.7	0.20	"	5.33	0.085	"	6.0	0.04

From Fig. 2.10, the maximum value of the mean vertical force in 1m high waves is approximately 23 Tf, in waves of length about 70m.

These results are for a fixed cylinder in small amplitude waves. Ogilvie also gives results for a free neutrally buoyant cylinder. The forces in this case are <u>less</u> than for the fixed cylinder. It is reasonable to suppose that in the intermediate region, for a damped cylinder, the forces will also be less than for the fixed cylinder. This conclusion is reached despite the contrary results deduced for the oscillating forces from Morison's Equation, for which relative phases are important. In the present case phase is not important : only mean values are considered.

The restriction that Ogilvie's theory applies to small waves is crucial. The phenomenon that it describes can be explained in terms of the Bernoulli effect of relatively high velocity flows passing through the gap between the top of the cylinder and the water surface. This would create a low pressure compared with the low velocity region below the cylinder, hence a net upwards force. The fluid motion described by Ogilvie's theory will be distorted as the ratio of incident wave height to mean depth of submergence increases, in particular when the top of the cylinder is exposed. In these cases it seems reasonable to expect a less substantial upwards force than Ogilvie's work suggests, indeed Salter (Ref. 4) has shown by experiment that the <u>total</u> mean force may then have a net downwards direction.

In conclusion, therefore, the shortcomings of theory to describe the complex flow about cylinders with small submergence in real waves suggests that only experiments can give convincing answers as to whether vertical mean forces are important. On present information they are not.

2.3c Mean Second-order Horizontal Force on a Submerged Cylinder

Longuet-Higgins (Ref. 5) has shown that the mean second-order horizontal force F_H acting on a two-dimensional energy absorbing structure is :

$$\frac{1}{4}\, g(H/2)^2(1+r^2-t^2) \qquad N/m$$

where r and t are the reflection and transmissions coefficicients.

For an efficiently tuned cylinder, $r = t = 0$. For an inefficient cylinder absorbing no power, $r = 0$, $t = 1$, and $F_H = 0$.

Thus as a simple guide to the maximum horizontal mean second-order force we use $H^2/16$ Tf/metre, or $F_H = 3.1H^2$ tonnes on a 50m cylinder.

This formula must also be treated with caution as it too assumes small waves. Salter (Ref. 4) has shown experimentally that this force can actually reverse in sign causing the cylinder to drift into the waves. Longuet-Higgins has also verified this experimentally.

It is unlikely that the small size of this mean horizontal force will cause any significant permanent displacement of the power takeoff and spring arrangements. Although it is not clear whether the mean displacement is with or against the direction of the incident waves, the magnitude in small.

2.3d Currents

The effect of a net horizontal force on the cylinder, displacing its axis either forwards or backwards from its mean position, could increase the frequency with which either the fenders (cushions) come into effect or slack rodes occur. A sufficiently large net force could seriously disrupt the pumping action described in Section 2.1 and 2.2, especially in the larger waves. We have therefore been anxious to establish the magnitude of likely currents off S. Uist, most particularly the component of these normal to the axis of a line of cylinders lying roughly parallel to the coast (in an average depth of, say, 40-60m).

The available data on currents in that sea area appears to be very limited. Admiralty charts show a maximum (spring) current of 2 knots roughly along the line foreseen for the devices, and we assume that this was recorded near the surface. If it occurred normal to any cylinder, say at the south end of the line as the tide flooded in to the east of S. Uist, the drag force on the cylinder would be about 30 Tf, or about 1% of the peak wave-induced force on the cylinder.

However, informal advise from Mr. Abdi (BP) cautions that information regarding currents given on charts can be misleading, and that it is not uncommon in local sea areas for currents during spring flood tides to exceed the chartered figures by a factor of two or three. This would suggest that the mean drag on cylinders exposed to such currents could reach up to 10% of the peak wave force. Although this would not be critical to the performance of the power takeoff device as at present envisaged, it is a sufficiently large unknown factor to require considerable further investigation and field measurement. Section 2.3e below emphasises the importance of taking this step as a matter of some urgency.

It is also worth noting that the sandy pockets infilling irregularities in the surface of the gneiss to the west of S. Uist generally show no sign of being regularly worked by currents. This endorses the records presented on the Admiralty charts. It does not mean, however, that locally strong currents do not occur in this area.

2.3e Combined Loading due to Waves and Currents

The kinematic components of waves and currents cannot be added vertorially to give the resultant fluid motion. As Refs. 6 and 7 make clear, this non-linear interaction can often give larger kinematic values than linear addition would suggest (e.g., as advocated in Ref. 1, p.34), and a 10% error in this calculation will appear as a 21% underestimate of the additional forces due to currents.

It was noted in Section 2.3d that the currents off S. Uist reported on the Admiralty charts are comparatively small and are certainly much less than will occur at the surface in high waves. Beneath the surface, however, the wave velocities rapidly decay whereas it is probable that, at least over the depth occupied by the cylinders, the currents will remain similarly strong (though there does not seem to be information to either confirm or deny this). It is therefore probable that estimates of wave forces made on components of the device located at more than 5m below the surface could be low because of the effect of currents, and that further calculations made on the assumption of linear superposition of waves and current velocity vectors will also be low as the magnitude of the current approaches and exceeds the orbital velocity due to wave motion.

In the circumstances it would be prudent to assume that forces calculated from wave action could be low, but it will not be until appropriate field data are collected that more precise calculations can be made. These data will also be needed for tank tests of all wave energy devices.

2.4 Principal Cylinder Dimensions

In our October 1979 Report we identified various reasons for favouring a cylinder 50m long and 12m diameter. Principal among these was the stability of the 4:1 aspect ratio (length:diameter) cylinder in oblique and irregular seas, in contrast to 6:1 and 2:1 shapes. Until further tests are carried out we have no means of judging the merits of, say, 5:1 and 3:1 relative to 4:1. The diameter of the cylinder, at 12m, is a direct compromise between a larger and more expensive cylinder that would experience much higher inertia wave forces, hence would require stronger moorings and anchors, and a smaller, cheaper, but less efficient cylinder. The optimum choice will be studied in Phase 4 of this investigation of the cylinder device, starting in April 1980.

At the time of writing there are therefore no clear reasons for recommending any move away from the cylinder dimensions given above. The argument made on structural grounds for using slightly domed ends to the cylinder may have to be slightly modified if, as seems likely upon closer inspection of its hydrodynamic performance, it is shown that vortex shedding, hence energy losses, occur from the still sharp edges. Softening these edges to a radius of 0.2m should overcome this problem.

The other leading dimensions of the device are the water depth in which it is located and its submergence. The choice of each is principally a compromise between the economics of capturing more energy by being in deeper water and having less submergence, but by so doing having to withstand greater wave forces. The optimum choice must again await the outcome of the further tank tests and related studies planned for Phase 4. However, some observations relevant to energy capture in deeper water and with less submergence are made in Sections 8.1 and 8.5 of the present Report.

2.5 Spacing between Cylinders

The minimum value of this parameter is determined by the need to ensure that the cylinders never touch each other. As important as this is, the primary consideration determining the specing may turn out to relate more to the process of lateral energy capture, whereby each cylinder harnesses energy from beyond its own projected length.

Theoretical studies of lateral capture efficiency recently made at Bristol University have related the maximum capture width of each of a finite number of equally spaced bodies (not necessarily cylinders) to that of an isolated body. By denoting this ratio by q, a value of $q > 1$ is clearly desirable. Theory suggests that, for five point absorbers spaced at about 0.8 of a wavelength, $q > 2$. As the number of bodies increases the optimum spacing also increases to a maximum value of one wavelength (L). On the other hand, with only 2 bodies the maximum value reached by $q \simeq 1.6$ when the spacing is about 0.6L.

Thus an infinite line of equally spaced bodies is theoretically capable of absorbing 100% of the incident energy flux providing only that their spacing is not more than a wavelength. For small bodies this will clearly mean that their amplitudes of motion must be large, reducing to reasonable size as their spacing is reduced.

These conclusions have recently been confirmed experimentally by Count et al in the Edinburgh wide tank.

If the bodies are replaced by cylinders it is not expected that the spacing requirements will change appreciably though the amplitude of the motion will be reduced. In a typical annual array of sea spectra it remains to select an optimum spacing according to the wavelength below which more energy is lost because the cylinders are too far apart, and above which they are unnecessarily close together. Although further theoretical and experimental work relevant to Phase 4 is needed to resolve this question, it now seems certain that the minimum spacing of 10m recommended in our October 1979 Report as necessary to avoid clashing is well below the optimum selected on the basis of lateral energy capture, for which perhaps 0.6L between the centres of adjacent cylindrical bodies is appropriate. If the threshold between a lower capture efficiency for shorter waves and a less than optimum spacing for higher waves is, for the present purposes of demonstration, taken to occur at L = 150m (T = 9.8 secs), the clearance between adjacent cylinders would be 40m. This means that only about half the length of a continuous device is needed to harness the bulk of the incident energy, though the optimum ratio of cylinder length to gap width may be shown from the Phase 4 study to be less than this. However, it is very important to ensure that each cylinder remains stable and performs efficiently in mixed and oblique seas, and it was for this reason that devices with an aspect ratio of only 2:1 were rejected (this was concluded from isolated cylinder tests).

On the experimental evidence so far available there can be no risk of clashing between cylinders that are 40m apart even if their optimum water depth turns out to be 100m rather than the 40-60m range at present regarded as preferable (Section 8.5).

2.6 Effects of Marine Growths on Cylinder Behaviour

It is too early to draw even provisional conclusions on this important issue though studies of the subject pertinent to the cylinder device have been put in hand. The present evidence suggests that the most serious problem will concern the use of seawater as the power transmission medium in the high pressure pipelines linking the pumps at each cylinder to the turbines. The available evidence on this particular problem is summarised in Section 6.4.

2.7 Device Behaviour in the Event of Component Failure

2.7a The Cylinder

This is internally divided into three compartments. If an end compartment floods, some buoyancy is lost at that end, but inertia is increased. Smaller orbits and less conversion of energy would result until the source of the leak was repaired and the water pumped out.

If two compartments flood then the cylinder sinks to the seabed where it could roll under the relatively small influence of the currents and residual wave action at that depth. Within its tethered reach this could cause considerable damage to sea bed mounted components.

2.7b The Corner Rode Assembly

Failure of a fore corner rode allows that end of the cylinder to rise and break the surface until its positive buoyancy is lost. It will then either drift downwave or upwave according to the direction in which the resultant force acts (see Section 2.3c - the presence of even a small current would probably determine this direction absolutely). The secured end of the cylinder would be slightly lowered by the drift force on the partially freed end, thereby slightly reducing the wave forces on that end. The unit would probably survive and would not damage the adjacent units because of the size of the gap between them (Section 2.5), but it remains for experiments to show the magnitude of the wave forces on the cylinder that must be carried by the aft rode at that end of the cylinder at which the fore rode failed before its chances of surviving can be assessed. If it also fails the cylinder will swing to be aligned generally into the waves, in which position the forces on it should be within the capacity of the remaining pair of rodes to contain. Experiments are also needed to check this.

Should an aft rode fail first the magnitude of the forces imposed on its neighbouring fore rode will again depend on the net direction of the wave drift and current action on the cylinder. As in the above case of primary failure of a fore rode, experiments in combined waves and currents are needed to demonstrate the circumstances that will then occur.

Should experiments show that rode failure causes excessive forces on other rodes such as to cause progressive failure, there are various methods available to safeguard the position, e.g., reduced rode design stresses or redundant rodes.

2.7c The Power Takeoff Unit

The design presented in our October 1979 Report included two pumps and four hydraulic springs per corner. The failure of any one of these components will put eccentric loads on the remainder. If the unit jams in its extended position, that end of the cylinder will be exposed to more severe wave action. On the downstroke the rode would go slack, and the resulting snatch loads on the upstroke could be severe. If this force either fractures the power takeoff unit or breaks the rode, the circumstances outlined in 2.7b above will then follow.

2.7d The Feeder Pipes from the Pumps to the Bus Main

Failure of a feeder means that the connected pumps lose their back pressure. The power takeoff units would still be symmetrically loaded but they would offer less restraint to the corner of the cylinder connected to them. This would magnify the orbit of that corner of the cylinder and impose greater loads on the springs at that corner. It is conceivable that progressive failure of the power takeoff unit would result, perhaps leading to failure of the rode (as in 2.7c above).

2.7e The Bus Main

This is fed from several cylinders. If it fails, the loss of back pressure could lead to the progressive failure of all connected power takeoff units, each producing the possible consequences for its cylinder as outlined in 2.7d above.

For the design reported in Oct. '79 this would result in the loss of output of eight devices with a nominal rating of 16 MW.

2.7f The Anchorages

It is envisaged that either tension piles or gravity anchors will be used, depending on the nature of the sea floor. If a tension pile fails the result would be similar to that foreseen in 2.7b above for failure of a rode. Failure of a gravity anchor, other than just dragging, could lead to two adjacent cylinder ends becoming under-restrained whereupon they would swing and ride more easily in a position skewed to their original alignment. This movement would probably fracture the associated feeder pipes (see 2.7d above), but would tend to lower the cylinders in the water and hence, together with the skewing, reduce the forces on them. The consequences of dragging a gravity anchor could be more serious if it also fractures the feeder pipes but without moving the cylinders significantly off station.

2.7g Conclusions about Failure Modes

Failure modes as described above can and should be designed out. The primary task is to recognise and evaluate them. The cylinder device is principally loaded by wave inertia, which is produced by and is linear with wave particle acceleration. Design is therefore relatively straightforward though it requires that the extreme and fatigue loading conditions at the location in question are specified according to a design life for the structure. A rigorous specification of wave heights, associated wave periods and number of occurrences in, say, a fifty year period at that location must therefore be made. For S. Uist the annual scatter diagrams mark an important step towards this specification.

The consolidated and optimised design concept foreseen as the main objective for Phase 4 of this study of the cylinder device will use such a specification to explore solutions to each of the possible failure modes described in this Section, in conjunction with representative experiments.

Planned maintenance procedure is also vital in minimising failure, and this will be considered in detail during the next phase.

2.8 New Concepts for Cylinder Mountings

We have continued to look critically at each of the main features of the reference design presented in our October 1979 Report; this had emerged by a process of logical selection from the wide variety of possible systems outlined in our June 1979 Report using the information then available to us. With additional experience and advice we have been reappraising this design, a process that will continue for as long as we feel that sensible improvements can be made. A compromise between economy, efficiency, reliability and credibility must be secured.

The use of stud link chain for the mooring rodes of our reference design has been criticised on grounds of reliability though it is an economic solution in which we continue to place confidence. However, as with all other components of the device, we have considered other ways in which its functions could be achieved. An account of these is given in Section 3.2.

The whole question of power transmission from the cylinder by linkage to conversion units on the sea floor has also been reviewed. Fig. 6.11 of our June 1979 Report proposed the use of fixed towers with power conversion adjacent to the bearings between the cylinder and the towers possibly using units like those already proposed at the sea floor. Although more expensive than rodes and exposed to wave forces should they project too close to the water surface, such towers should be more reliable in use than rodes. However, this does not mean that rodes cannot be made adequately reliable (see Section 3.3) and there is the problem with rigid frames of locating them accurately in position, in hostile conditions, to take the mountings of the cylinders.

SECTION 3

MOORINGS

3.1 Analytical Specification of Mooring Forces

The step-by-step non-linear analysis of wave-induced forces presented in Section 2.2 was devised to take account of the wave/cylinder/mooring/pump/cushion interaction. It has only recently been completed and has so far been applied to the single case of waves of height 9m, period 7.55s, length 88.7m and steepness 1/9.87.

Because of its steepness, the theoretical motions and forces produced by this wave were particularly vigorous. The maximum calculated force was provisionally shown to be nearly 8600 tonnes per side (4300 tonnes per corner), and it assumed that the piston and spring over-ran the top stop by some 0.6m at increasing stiffness.

The peak mooring force had previously been calculated to be 3,250 tonnes per side (Fig. 3.7, Oct '79 Report). This applied to a fixed cylinder, 3m shorter, in waves of height 16m, period 11.55s, length 204m, and steepness 1/12.75.

Clearly the new results must be interpreted with caution. The forces suggested in this one test case are greatly in excess of those calculated by a simpler analytical procedure and checked by experiment. However neither that simpler analysis nor the experiments included the proper representation of the performance of the pumps, springs and cushions, all of which are included in the new method of non-linear analysis.

If the trend towards larger forces and motions identified by the new analysis is confirmed by the more comprehensive application of this method and by experiments using similarly representative models, it will mean redesigning many components of the device. However it is much too early to assume that this will be necessary and in any case neither that re-design nor any other hardening of concepts can sensibly be undertaken without a closer specification of design wave steepness (Section 2.3a).

An immediate task will therefore be to review the details of the new method of analysis with a view to programming it for use on a digital computer. In that form it may then be used conveniently to explore such design aspects of the device as the effect on conversion efficiency of limiting the motion of the rodes, the forces then imposed on the motion limiting units, and the occurrence and consequences for the whole system of slack and snatch in the rodes. This model of the dynamic performance of the device will indicate the forces imposed throughout it for a wide range of representative wave heights and lengths. Fatigue spectra will be deduced from these results and applied to determine the design requirements of components sensitive to this form of loading.

The accuracy of the model will first be checked against data from the representative tank tests scheduled to start early in Phase 4 of this investigation. The model and the tank tests will then be operated in parallel so as to make the maximum use of each.

3.2 Other Mooring Methods

Alternatives to the use of chains for rodes have been given attention. This matter is seen as inter-related with the use and design of pin connections. Notwithstanding the comments and criticisms that have been voiced in some quarters regarding pins and chains, it should be realised that these components are currently used for similar duties in marine environments. An example would be the Single Anchor Leg Mooring system. The maximum duty required as a cylinder mooring was previously calculated to be a tension of up to 2000 tonnes whereas North Sea practice already extends to 7000 tonnes load on a pin joint.

Chains can obviously be replaced by other linkage systems having the necessary articulation, such as tubes with carefully designed joints (see Section 3.2b below). These joints could either by metal hinge or ball and socket type or could possibly be a steel/rubber sandwich construction. We have looked at the possibility of using prestressed concrete to see if any benefit can be gained in greater fatigue strength (see Section 3.2a below).

Fatigue is one of the main problems with the mooring rode and there is a considerable lack of information on the fatigue characteristics of chains and parafil ropes, which could be another option though in the latter case end terminations would also present a major problem. These matters have been discussed with TAG 4.

3.2a Composite Prestressed Cable

i) Outline of Concept

A common feature of all catenary moored floating systems is that the mooring cables or tethers are subject to a varying tension ranging from the maximum service load to a value approaching zero. To obtain a satisfactory working life it is necessary to operate the cables at a suitably low stress factor in order to keep within the endurance envelope of the material. Efficiency would evidently be improved if the stress range could be reduced under conditions in which a higher mean stress would be acceptable. The possibility of achieving these conditions for a wave energy device has been investigated using the concept of a composite cable consisting typically of a tendon surrounded by a continuous concrete sleeve, the combination being prestressed prior to placing in service.

By specifying the relative cross sectional areas of the compression and tension components it is possible within limits to transfer the problem of stress fluctuation from the tension to the compression member. The possibility of using articulating joints with convex mating surfaces between each element of the compression annulus has been examined but it has not yet been possible to find solutions to the obvious practical problems arising from the very high contact stresses involved.

It is therefore concluded that the necessary curvatures must be obtained by elastic flexure within the limits dictated by combined flexures and direct stress levels and when this is insufficient, the cable must be fabricated in discreet lengths, of perhaps 30 metres, linked together by universal hinge joints.

ii) Behaviour under Direct Tension

The distribution of forces between the tendon and the compression annulus, assuming completely elastic behaviour, is illustrated in Fig. 3.1. With zero external force, the tension and compression forces are zero (Point A).

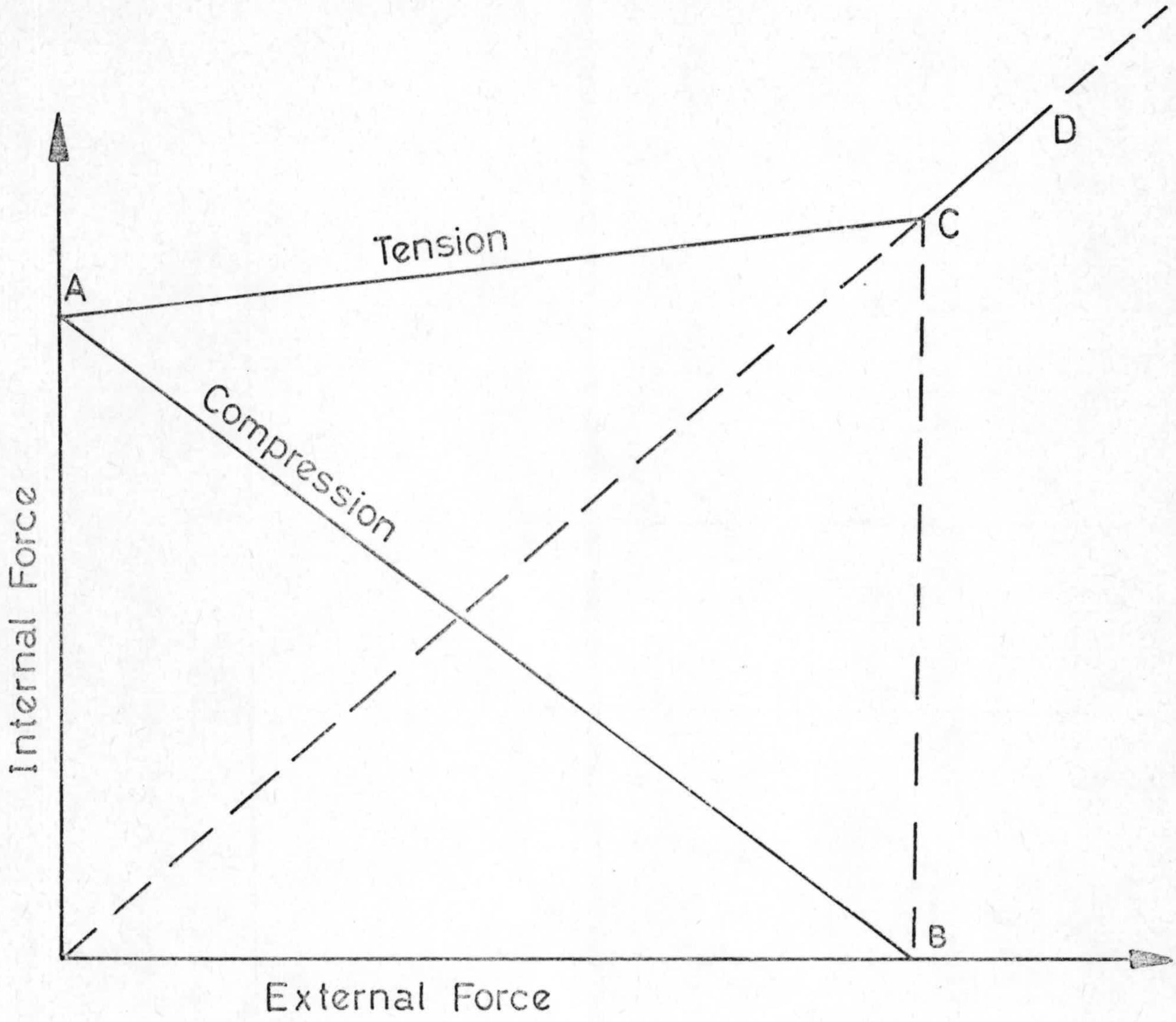
When a progressively increasing external force is applied the tension in the tendon increases along AC and the compression in the annulus decreases until at Point B, the annulus is fully decompressed. Further increases in external force are then carried entirely by the tendon (CD).

The extent to which load variation is absorbed by the compression annulus in preference to the tendon is determined by the relative stiffness of the two components and is indicated by the relative slopes of lines AC and AB. In practice, the position CD of the curve, implying joint opening in the compression member, would be inadmissible and point C would be set slightly above the maximum design load for the cable.

As an example, consider a cable designed for a maximum load of 500 tonnes having an outside diameter of 126mm. The load variation can be reduced to 20% by providing a compression annulus 380mm dia.

iii) Properties in Flexure

Unlike a chain or rope, a prestressed member must transmit compression as well as tension. It follows that the composite will have a relatively high rigidity which severely limits its compliance and restricts its ability to accommodate the varying geometry of a floating system by elastic flexure alone. In this respect the greater the efficiency of the system in reducing load variations in the tendon the greater will be its rigidity and the less will it resemble a cable in its behaviour.


D
C
Tension
A
Compression
Internal Force
B
External Force

iv) Provisional Assessment of Concept

From the work done it is evident that while the concept might have some advantages in relatively deep water, the considerable practical problems involving both mechanical performance and material requirements, including corrosion considerations, are a deterrent to applying the concept to wave energy devices.

3.2b Steel Tubes and Rods

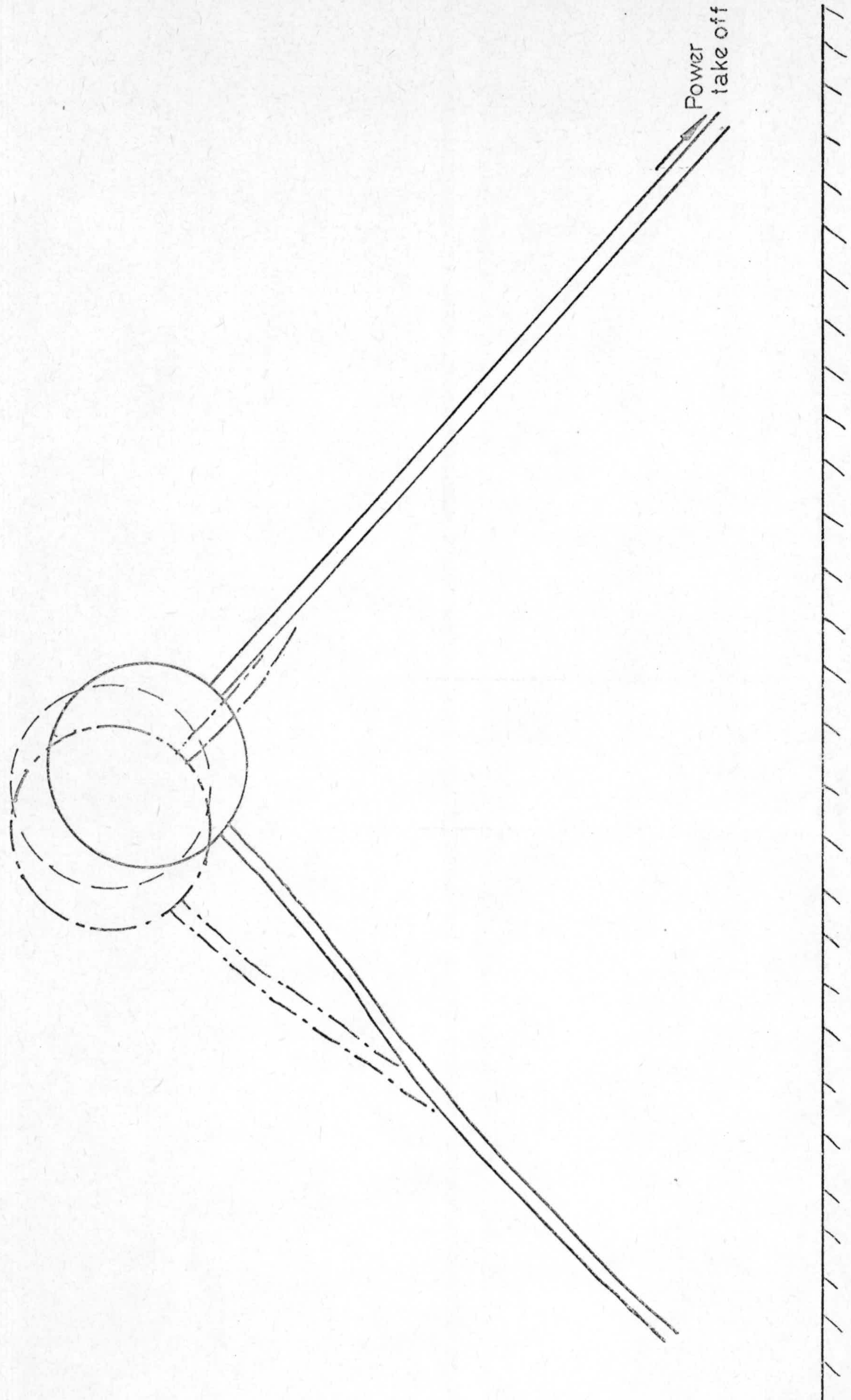
These options to chains have yet to be evaluated in detail though they appear to be acceptable in principle. The use of 3-5 engineered joints per rode will distribute the rotation to be carried at any joint, and the use of tubes or rods will minimise the problems of surface corrosion and erosion, and if needed will make protection easier to apply and maintain.

The use of articulated joints will, as with chains, prevent the tube or rod members from going into compression. An alternative arrangement would be to use continuous rods or tubes, built into rigid mountings on the cylinder and constrained by bearings to move to and fro in a constant direction at the sea bed (akin to a cantilever but with the free end built into a moving member that allows it some rotation at that end).

Fig. 3.2 shows the limits to the motion of a 50m long tube attached to a cylinder orbiting through 3m. In addition to transmitting one component of the cylinder's motion to activate the power takeoff units, the bending of each tube will serve, at least in part, as the spring force required to tune the motion of the other tube. A study is now being made of this option, which combines simplicity with a technique for avoiding the use of pins and possibly also springs of, say, the oil/nitrogen type hitherto advocated.

The particular disadvantages of this arrangement are the bottom bearing and the need to ensure that the tube is the correct length to suit the foundation level provided and the submergence depth chosen above the cylinder.

FIG 3.2



3.3 Fatigue Life of Moorings

Reference was made in Section 2.7g to our continuing need for a clear specification of the extreme and fatigue loading forces on the cylinder off S. Uist. Hitherto we have followed the advice given to us about chains that if the maximum load is kept within 60% of proof load the rodes would have a long fatigue life. We note that this information will be updated by Mr. Arne Berg of Det Norske Veritas in his Paper to the Offshore Technology Conference, Houston, May 1980. However, the new information will only be as useful as the confidence placed in the statements that remain to be made about the more severe wave loading conditions.

3.4 Alternative Mooring/Power Takeoff Arrangements

The reference design presented in our October 1979 Report used rodes connected to the ends of the cylinder and which lay in a plane normal to the axis of the cylinder.

In-board rode connections to the cylinder would significantly reduce the longitudinal bending and shear stresses on the cylinder walls, which would slightly reduce costs. The consequent cylinder motions in oblique seas must be studied by model tests. The motion and forces on the system could be more vigorous, and if the cylinder is less stable its capture efficiency might be reduced. No structural advantage would be gained by moving the rodes in-board by more than one-fifth of the length of the cylinder.

Fig. 3.3 shows inward and outward splay to the cables. In both cases the stability of the cylinder should be improved though power may be transmitted less efficiently to the sea floor. The rodes would also be marginally longer. Most important may be the economic savings offered by simplified anchoring arrangements, thereby countering one of the arguments against increased spacing between cylinders (Section 2.5). With greater water depth and in-board connection points onto the cylinder, centralised anchors for each cylinder could, for example, be connected up with relatively little splay to the rodes.

FIG 3.3

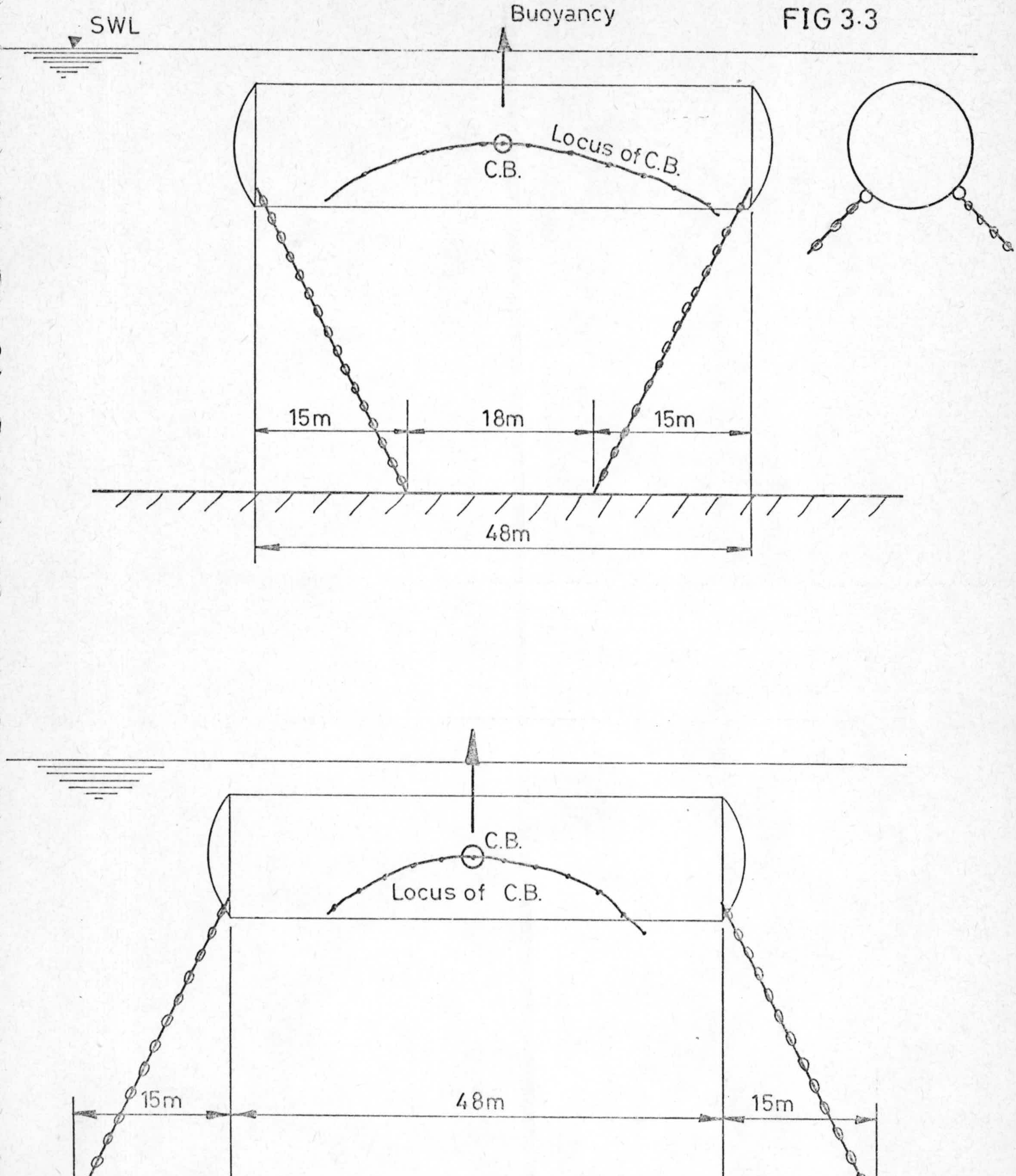

SWL
Buoyancy
Locus of C.B.
C.B.
15m
18m
15m
48m
C.B.
Locus of C.B.
15m
48m
15m

Fig. 6.2 of our June 1979 Report conjectured on the possibility of using cams to change the direction of the transmitted force, though for our reference design we decided to adopt an in-line arrangement between the rodes and power takeoff unit (Fig. 6.4, Oct. '79 Report). However, in the 42m depth adopted for that design, the overall length of the power takeoff unit becomes a substantial proportion of the distance from foundation to anchor point, especially if gravity anchors are used.

We have therefore considered the option of setting the power takeoff unit in a horizontal position, and connecting the rodes to it via cams or cranks. In this position the unit would need two tension piles to stabilise it, together with its cam or crank, rather than one for the 'in-line' arrangement. This would therefore require use of a jigged completion assembly to install the pair of tension piles, and would cause increased load on the pile close to the cam or crank. The case against a horizontally-mounted unit is much less strong when it is used in conjunction with a gravity anchor, to which it can either be connected before submergence or arrangements made to ensure that this can readily be done in-situ.

On present evidence, however, there is no outstanding reason for rejecting or substantially changing the reference design layout previously proposed, though the case for introducing amendments will be kept under continual review until the optimum water depth (see Section 8.5), mooring method (Section 3.2) and power takeoff units (Section 5) have been chosen.

3.5 Marine Growths on the Moorings

There appears to be little doubt that the moorings will quickly be colonised by a variety of marine growths including mussels (bi-valve molluscs) and various crustaceans (a more complete account is given in Section 6.4). It is unlikely that this colonisation will interfere with the intended performance of the moorings. The current-induced drag force on the chains is well under 1% of the average tension they must carry, and even if this is increased to 1 or 2% by the links becoming overgrown the main function of the chains will not be impaired. The extra weight on the chains will be very small.

Neither the natural form of chains nor the profile they will present to the currents when encrusted is likely to allow coherent vortex shedding to develop, hence flow-induced vibrations of the rodes can be dismissed. The situation with moorings made from tubes or rods could be different, especially before they become roughened by marine growths. They would often operate within the transitional range of Reynolds numbers (2×10^5 - 5×10^5), when the vortex frequency would lie in the range 0.2 - 1.0 Hz approximately. If rods or tubes are preferred to chains, the significance of fatigue loading at these frequencies on the life of the rodes should not be overlooked.

SECTION 4

ANCHORAGES AND SEABED CONDITIONS

4.1 Rock Types, Integrity and Surface Form

It is generally maintained that Lewisian gneiss forms the sea floor off most of the west coast of S. Uist to a depth of at least 200m. Numerous pockets of sand, that are usually stable and which sometimes include a small mud fraction, occur throughout this area, and these may occupy a depth in excess of 1m.

The gneiss is known to be faulted as part of the substantial folding that extends throughout north-west Scotland. It may therefore be expected that this is a fractured water-bearing structure that locally may extend to considerable depths. The surface texture is reported to be rough but the significance of this can only properly be interpreted in the context of the installations that have to be made on it. Echo sounding evidence can easily give a misleading picture of roughness. The peaks estimated by Kenyon and Pelton (Ref. 8) to be over 1m high every 30m imply that bed gradients are on average 3-4%, though it is probable that locally steeper slopes will rise above a generally flatter topography extending over much of the area.

For the cylinder device it is necessary to translate this information, according to the chosen average water depth for the devices, into terms that allow :

a) an acceptable surface slope and rockhead elevation to be chosen within a prescribed distance of the preferred conditions specified;
b) the problems of gravity anchors to be quantified;
c) the problems of running large hydraulic mains, of about 1m diameter, between devices to be estimated with a view to predicting the need to mount these on levelled-in piers or in prepared troughs to minimise bending moments.

4.2 Anchorage Methods, Installation Procedures and Costs

At the Maidenhead Workshop, December 1979, considerable discrepancies between various device teams' cost estimates for anchor piles became apparent. We were at the expensive end whilst Wavepower Ltd. were considerably cheaper. They had consulted Wimpey whilst we had an estimate from Raymond International which we had discussed briefly with Babtie, Shaw and Morton (whose recent experience on the Kielder tunnel is reported in Ref. 9), and with BP.

Since Maidenhead these major differences have been considered further both with TAG 4 and WPL. The following conclusions have emerged.

1. The seabed off the Outer Hebrides in 40 to 50m of water is believed to be mainly gneiss (a hard rock) with an irregular profile. Unfortunately, there seems to be little firm information on the degree of irregularity in local areas. Further information on this will be necessary before either anchor piles or the methods of installation can be properly designed.

2. Drilling into this rock will have to be done from a jack-up platform or barge for any size of pile.

3. Two factors have the main effects on the cost of anchor pile construction :
   (a) speed of drilling into the rock and;
   (b) the amount of down time due to sea conditions.

4. Though there are differing opinions on the possible speed of drilling, which must be resolved, the effect of this on costs can be minimised if the jack-up barge can be designed to drill more than one anchorage at a time. It is believed that up to 8 concurrent drillings may be possible for the cylinder device with the spacings envisaged in our reference design.

5. The amount of downtime is controlled by the sea conditions in which the jack-up barge can move from one location to another. If severe hold-ups are to be avoided, the barge must be movable in wave heights of up to 3 metres. It is thought that there is no suitable barge for this duty in existence at present but there seems no reason to believe that one cannot be designed. The barge must be capable of spudding to the seabed at the right location in these heavy seas. In view of the irregular seabed anticipated, it may be that the barge will have to place some form of mattress on the seabed prior to lowering the legs.

6. Though the cost of a purpose built jack-up drilling platform will be high it will have a dedicated use for the 10 years construction at present envisaged for a 2GW power station and hence the cost of the platform per device will be modest.

Arrangements will be made in the optimisation phase of the study to clarify and quantify the above points so that the feasibility and costs of anchor piling may be determined within reasonable limits bearing in mind the geological information available.

SECTION 5

POWER TAKEOFF UNIT

5.1 Introduction

The components that made up the power takeoff unit of the reference design, and which were presented in Section 5 of our October 1979 Report, have been widely discussed and considered further by us. We have also continued our studies of other ways of performing the various functions required of the power takeoff unit, and have given attention to the possible choices of materials for components fulfilling these functions.

Some provisional observations on the outcome of these continuing enquiries are presented in this Section. They will be firmed up in Phase 4 as further information becomes available on;

a) the actual duties required of the power takeoff unit within the overall system;
b) the performance and probable cost of other types of pump and spring;
c) the probable effects of marine growths and how these should influence the design of the device (Section 6.4);
d) the interaction of metals with seawater (Section 5.5).

5.2 Rating of Power Takeoff Unit

We are conscious that the power takeoff unit presented in Section 5 of our October 1979 Report was designed without detailed reference to the functions it will have to perform in association with a 50m long, 12m diameter cylinder, although the component sizes then suggested should form a good first estimate of the requirements of the device.

The rating of each power takeoff unit had to be chosen on scant experimental and wave climate evidence, but that was sufficient to give first order estimates of sizes, availability of components, and costs. A more precise rating will be possible after further model test results and wave data become available. At that stage it is also expected that the optimum length, diameter, submergence, spacing and water depth for the cylinder will be more closely fixed so that, in conjunction with wave climate and device performance data, a comprehensive specification for the functions required of the power takeoff unit can be drawn up and matched by the most appropriate systems then available. This will indicate the number of cylinders that will be needed for a 2GW station, the sea length that these will occupy, and the mean annual output they will deliver to the network.

5.3 Horizontally Mounted Power Takeoff

This possible arrangement was referred to in Section 3.4 for the way in which it would affect the moorings and anchors.

Such a scheme would involve the use of cranks or cams with very highly loaded limited motion bearings. The wear rate in these conditions of corrosion and possible abrasion (see Section 5.5) is likely to be very high for metals. Special composites and plastics are being studied but the component sizes required may be excessive. The load ratings of the bearings are directly related to the peak loadings, but, as implied in Section 5.2 above, the detailed evaluation of these is part of the experimental tests scheduled for Phase 4.

5.4 Other Pump and Spring Systems

Alternative pumps will be considered in detail if it is found from analysis and model testing that constant load pumping is not acceptable. A widespread survey of possible options is now well advanced and we will be intensifying our studies of the more promising of these as further information about each becomes available.

Regarding tube pumps, Wavepower Ltd. expect to have more information on pump efficiency, maximum pressure and fatigue life by Autumn 1980. The detailed device performance studies that we will shortly be commencing should, by that time, allow us to decide on our pumping requirements, and this will permit the device specific appraisal of the tube pump to be carried out.

We are also following up a range of pumps developed by BP (Research & Development) for use with sea water. These comprise various arrangements of concentric steel cylinders and rubber bushes. Some performance data are available but the size of the units and the pumping pressures required by BP's in-house needs are well below those preferred for the cylinder device. However, the principles of such a system are attractive and deserve further consideration. Like the tube pump, these devices could perform both the pumping and spring functions, as well as act as at least part of the rode.

One possible disadvantage of these pumps that has to be quantified is the amount of energy they may absorb internally, the heat this may generate, and the limited lifetime it may impose on critical components. In this respect the examinations we have made of rubber springs have indicated that, in our duty, they are likely to have high internal energy losses leading to overheating and uncertainty about the fatigue life of the rubber to metal bonds. In comparison, the spring system in our reference design employs a gas (nitrogen) for energy storage : with correct design the energy loss and consequent heat generation in this system should be small.

A more complex power takeoff unit than that proposed in our reference design is also under consideration. This uses oil hydraulics for both the pump and spring functions. It would include a control system to determine the proportions of the energy captured that are either to be stored or converted to electrical energy using a hydrostatic motor and generator. This system has the potential for including special tuning if that appears to be required to optimise output according to the performance characteristics achievable (see Section 8.1), and to reduce peak rode forces.

Other possible tuning systems are also under review. The case for including them in optimised designs of the power takeoff unit will depend on the improvements in overall device performance they are likely to offer, also the cost of including them and the reliability that can be placed in their performance in relatively inaccessible and inhospitable conditions.

5.5 Materials for Power Takeoff Units

Studies of materials are continuing with particular reference to the erosion, corrosion and reduced fatigue life of metals in sea water.

The corrosion rate of some stainless steels is highly dependent upon local fluid velocity, hence data on the magnitude and duration on the probable currents in the lower part of the water column are needed (see also Section 2.3d). The abrasive nature of the sea floor sand is also unknown, though, as mentioned in Section 4.1, this material is apparently stable at the depth of 40-60m of interest to us over at least much of the area to the west of S. Uist.

We suspect that much more information about the durability of a range of metals at depth off the Hebrides is in urgent need of assembly. The Harwell Materials Group may have this in hand.

5.6 Choice of Limits for Pumping Stroke

Fig. 2.2 of our October 1979 Report showed that the orbit of the cylinder when damped is less than that of the waves, and that the amplitude ratio (= cylinder ratio/wave height) appears to tend to a limiting value of not more than 4m in high waves.

The tank tests scheduled for Phase 4 of our study will clarify the details of this important relationship according to the leading dimensions chosen for the cylinder, the wave climate, and the characteristics and magnitude of the damping and springs used.

The diameter of the cylinder and its submergence will determine the forces imposed on it by the incident waves. If it is assumed that, without springs or dampers, the cylinder has an orbit equal to the orbit of the waves that, in its absence, would occur at the depth of its centre, the relationship between wave height and cylinder orbit would be as shown in Fig.5.1 For any combination of wavelength and wave height (hence wave steepness, s), the -axis of Fig. 5.1 shows the undamped orbital diameter of a 12m cylinder submerged by 3m below mean water level. The ratio of this orbit to the incident wave height is also shown on Fig. 5.1 It would appear that, for any wave steepness, the ratio of cylinder orbit to wave height increases with increasing wavelength (and hence with wave height).

The experimental relationship between cylinder orbit and wave height presented in Fig. 2.2 of our October 1979 Report is also shown in Fig. 5.1. In contrast to the observations reached above for an undamped cylinder, the orbital ratio now decreases from more than 0.8 to less than 0.4 as wave height increases to 10m. It follows that the total damping applied to the cylinder must be increasing with increasing wave height, and this must relate to the characteristic of the cylinder to shed an increasing proportion of the incident energy as wave height increases. Fig. 7.6 of our October 1979 Report gave experimental data for this relationship according to wave period, and Fig. 7.7 of that Report compared the annual wave height distribution of incident energy with the equivalent distribution of captured energy for a cylinder tuned to 8 seconds. The much more comprehensive tests planned for Phase 4 of this study (Appendix B) will explore the damped orbital relationship in more detail, especially for irregular waves. It is of great importance to the design of the power takeoff system.

Fig. 5.2 has been prepared from this information to show how much of the annual energy resource is captured by allowing the orbit of the cylinder to exceed any particular value. Present information does not permit us to estimate the capture efficiency when the cylinder orbit is additionally restrained by the action of cushions. That information will be collected during Phase 4 of this study to allow the optimum location and characteristic of cushions to be identified.

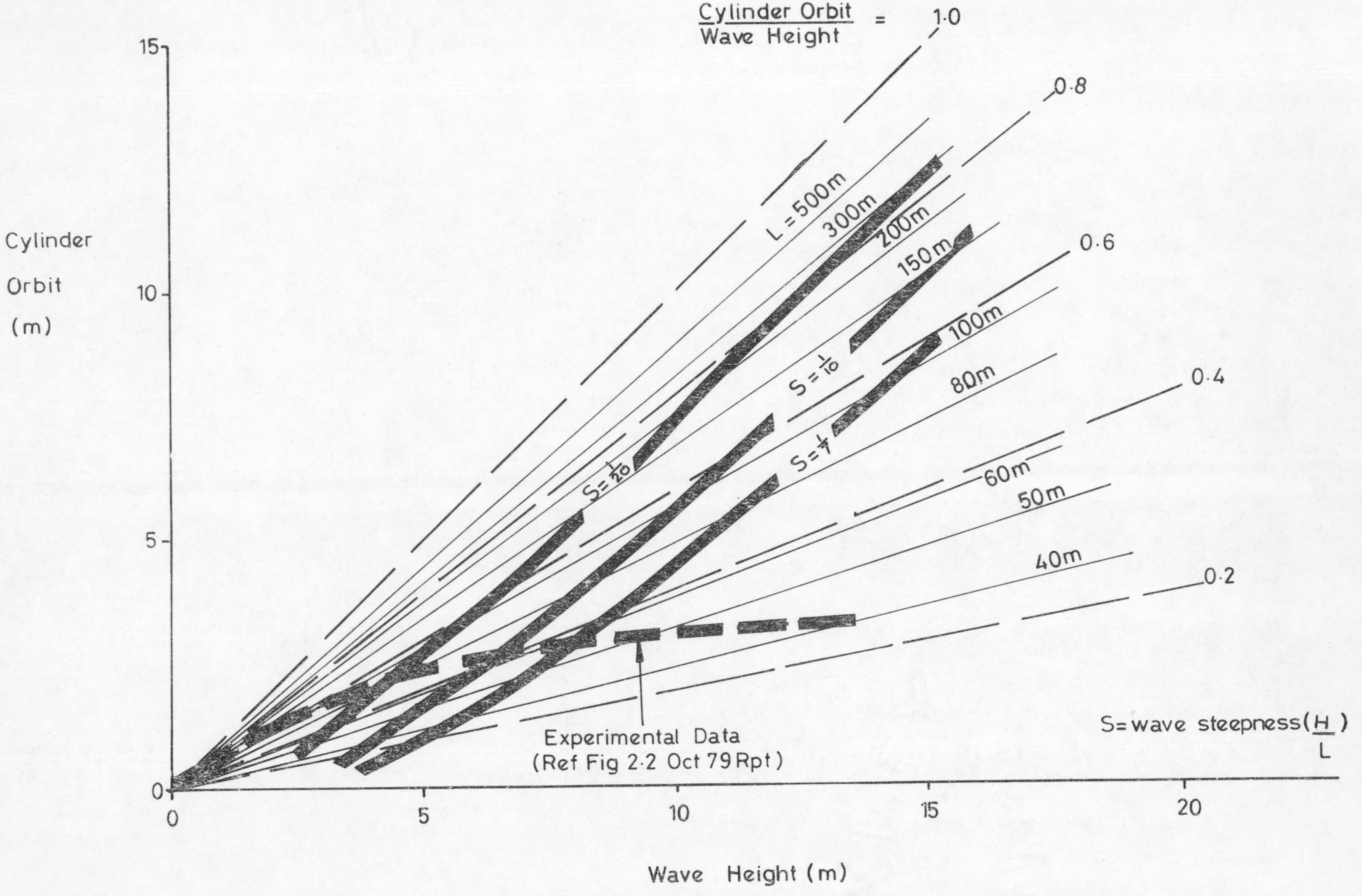

Cylinder Orbit / Wave Height = 1·0
0·8
0·6
0·4
0·2
Cylinder Orbit (m)
Wave Height (m)
0
5
10
15
20
L = 500m
300m
200m
150m
100m
80m
60m
50m
40m
S = 1/20
S = 1/10
S = 1/7
Experimental Data
(Ref Fig 2·2 Oct 79 Rpt)
S = wave steepness (H/L)

FIG 5.1

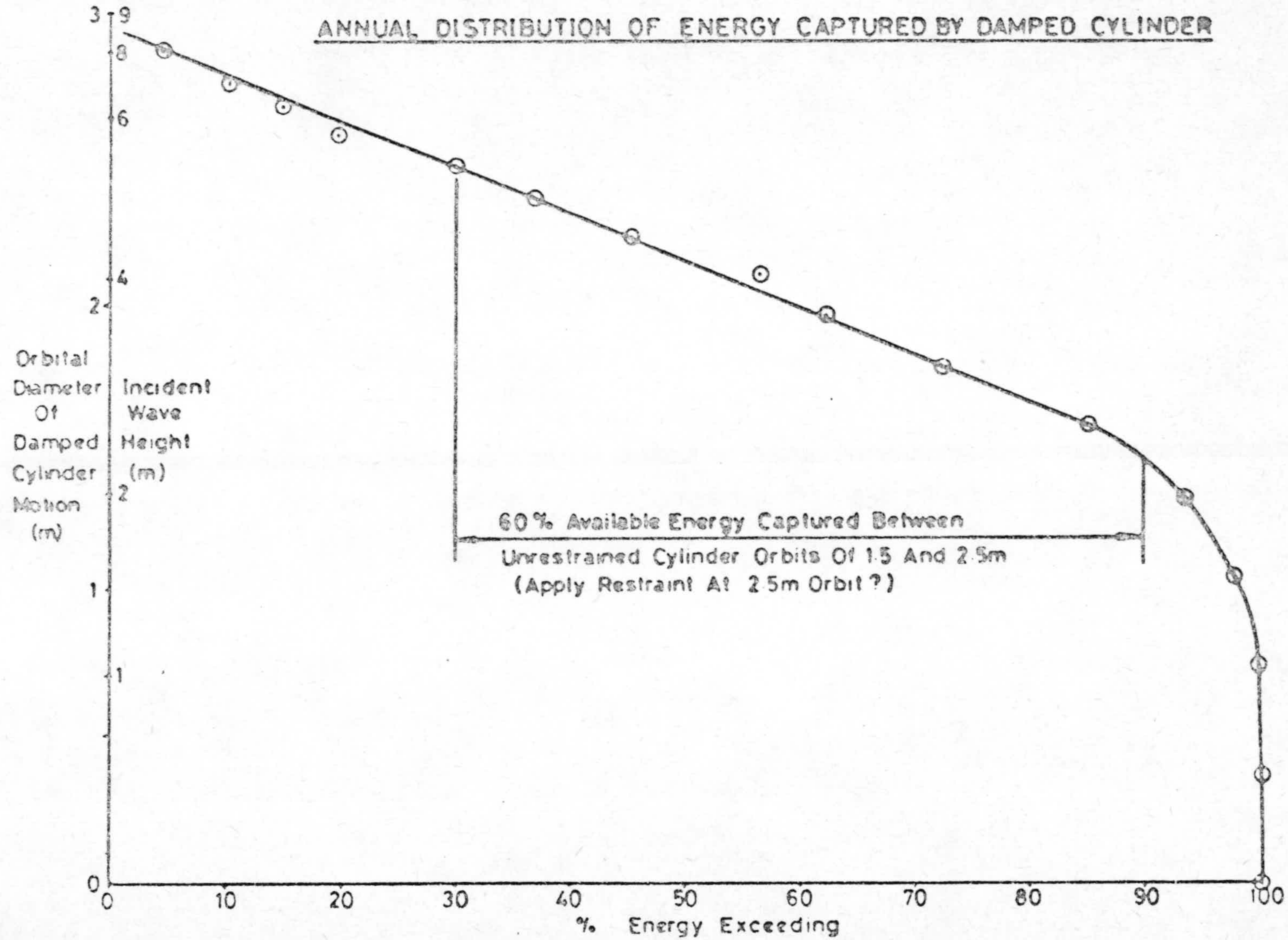

ANNUAL DISTRIBUTION OF ENERGY CAPTURED BY DAMPED CYLINDER
Orbital Diameter Of Damped Cylinder Motion (m)
Incident Wave Height (m)
60% Available Energy Captured Between Unrestrained Cylinder Orbits Of 1.5 And 2.5m (Apply Restraint At 2.5m Orbit ?)
% Energy Exceeding
0
10
20
30
40
50
60
70
80
90
100
1
2
3
4
6
8
9

FIG 5.2

SECTION 6

HYDRAULIC TRANSMISSION, TURBINES, LINK TO GRID

6.1 Introduction

The proposal in the reference design of our October 1979 Report included the seabed transmission of medium pressure sea water (600 - 700 p.s.i.) from power takeoff devices to converter towers where turbo-generators, located above sea level, convert the hydraulic power to electricity. The electrical power was then transmitted via submarine cable to shore. This design resulted in an excessive number of 1 metre diameter seabed bus mains and this proved to be a major cost centre in the design. Clearly, it is necessary to produce a more elegant and less expensive design. Various options are open :

1. to use higher pressure sea water and hence reduce the number of large bus mains;

2. to use high pressure fresh water closed circuit;

3. to use high pressure oil closed circuit;

4. to retain the use of medium pressure sea water but minimise the number of bus mains by using either :

   (a) large diameter pipes or submerged tube, both jointed on the seabed, or;
   (b) tunnels in the sea bed rock, or;
   (c) more smaller turbo-generators per 2GW station.

Option (1) will result in greater pipeline wear and cause complications with turbine design. Neither of these is insuperable and are being investigated. Options (2) and (3) are open to the objection of requiring return pipes thus losing at least some of the advantages of increasing the pressure. Option (4) is promising and will receive further study in Phase 4 of our work. A preliminary assessment has been made of the use of tunnels to define the possibilities : this is reported in Section 6.3.

6.2 Optimum Arrangement of Hydraulic Transmission and Turbines

For the reasons given in Section 6.1 above, it is important to identify the optimum balance between the number of devices that should supply each turbine and how that supply should be transmitted. While this cannot be decided without the performance and cost data that will be assembled in Phase 4 of this study, some first observations about the choice of hydraulic transmission arrangements can be made, irrespective of whether these turn out to be best done using, say, tunnels rather than pipes or submerged tubes.

The amount of energy transmitted by hydraulic system depends upon the power level adopted and the duration of the transmission. The power is determined by the operating head and by the discharge, whereas the transmission efficiency is determined by the flow velocity adopted, the distance involved, and the hydraulic roughness of the bus main.

Energy lost in transmission is equivalent to a reduction in the pressure head along the length of the main. This is given by

$$h_1 = \frac{\lambda M V^2}{2gD_o}$$

where λ = hydraulic roughness coefficient of the main
M = length of main
V = average velocity (=Q/A)
Q = discharge rate
A = cross-sectional area of flow in main (= $\pi D_o^2/4$)

The kinetic energy of the flow and the losses in bends and valves will be small compared with the above friction energy loss for the long mains required by the present application.

The effective head at the turbine will therefore be :

$$H_p - h_1$$

where H_p = pumping head chosen.

The power transmission is then given by

$$P = wQ(H_p - h_1)$$

where w = unit weight of the fluid used in the main. If, as hitherto generally assumed, this is sea water, it is very probable that deposits of some form will collect along the main (see Section 6.4 below). In this case λ will increase in time, and D_o (hence A) would decrease. For the same flow rate Q, the velocity V will increase. Each of these changes raises the value of h_1; hence the efficiency of the transmission, which is effectively $(H_p-h_1)/H_p$, will decrease significantly.

It is clearly advantageous to minimise losses as far as economics permit. This points to the use of the maximum feasible pressure head in the main. The power available for transmission will then be delivered by the minimum possible discharge. It then remains to select an acceptable diameter for the main, which will determine the velocity and hence h_1. Thus with high H_p, low Q and relatively large D_o, the ratio h_1/H_p can be reduced to small proportions. It can be reduced still further by reducing M, i.e., by installing more, smaller turbines so that each is fed by fewer devices. The balance will clearly depend on economic factors. It is unlikely that because a 200MW unit (turbine plus platform), for example, is much less expensive than 2 No. 100 MW units, the optimum solution will favour the use of short lengths of main between the devices and their turbine. As far as hydraulic transmission efficiency is concerned, this strengthens the case for high pumping pressures, to keep transmission losses to a minimum.

The optimum choice of H_p, M and D_o will depend upon the output characteristics of the power takeoff unit in response to the capture efficiency of the cylinder device over the range of wave conditions in which it will operate. Phase 4 of this study will provide this information. It will then be possible to select the optimum parameter values for the transmission main, bearing in mind that higher than average power output may be of less than corresponding electrical value to the system, hence to bias the design of the main in favour of transmitting them at high efficiency to the turbine may imply use of an unjustifiably expensive (large) main.

6.3 The Use of Tunnels as an Alternative to Pipelines

6.3a

Two possible schemes are considered here. Reference should be made to Fig.6.1. Each is based on the following common components :

a) 100 No. cylinders pumping sea water at high pressure to;

b) 100 No. downpipes, 0.3m diameter, and;

c) 600m long collecting tunnel, 2.5m diameter supplying;

d) Shaft from tunnel to seabed at the collecting tower.

Scheme 1 includes a tower 10km offshore founded on the seabed and equipped with say, a 200MW turbo-generator. Power is conveyed to shore by seabed cable.

Scheme 2 includes a single 3.5m diameter tunnel running 10km to shore, connecting the outputs of the 100 cylinders (or more) to on-shore turbo-generators. The tower founded on the seabed to serve as a working platform while these tunnels were drilled could subsequently be removed for use elsewhere.

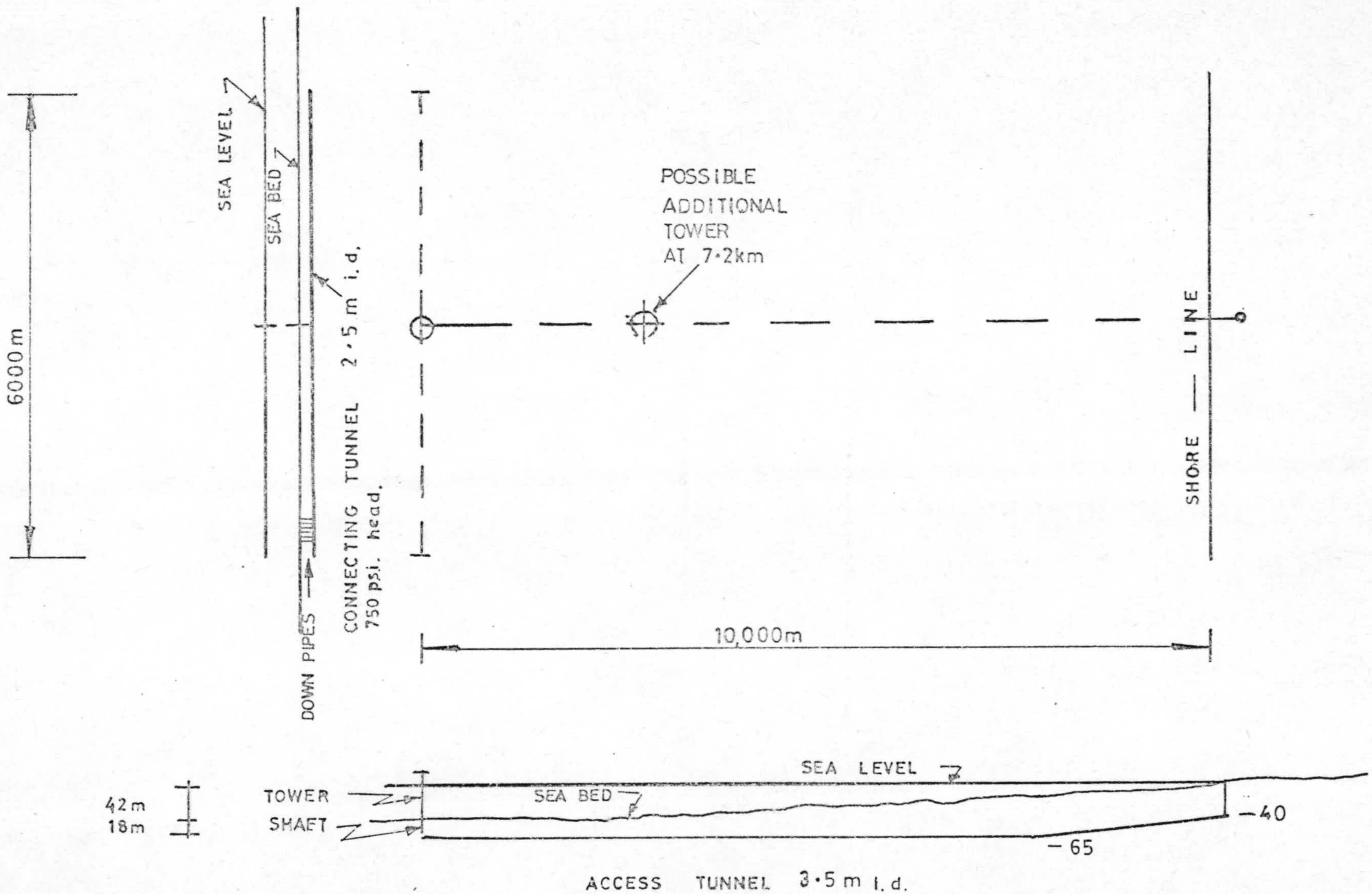
6.3b Consideration of the Problem

1. Tunnels

The main problem facing the tunnel scheme is the nature of the ground. It is believed to be gneiss which is expected to have a high compressive strength.

Its nature is, however, not known. In particular this is a matter of the joint pattern and the frequency of fissures that occur.

In order to avoid at least some of the weathered zone and the more open joints the assumption will be made that the invert to the



SCALE:- 1in = 2000metres

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.1

tunnel is at 18m below the seabed. This gives 15m of cover, approximately, and a total depth below sea level of 60m if the 42m water depth location is adopted. As a comparison it may be remembered that the Severn-Wye tunnel is 30m below bed level but was also very wet. If necessary the level of the wave energy tunnel may be lowered without a great deal of extra cost, though the pressures in it will increase pro rata if the turbine operating pressure is unchanged. In the case of the access tunnel the above still applies. In addition, to reduce the dangers of flooding, the gradient of the tunnel should be upwards from the shafts. However, it may be acceptable to reduce the depth of the land shaft and slope down at 1:100 for 2500m, from, say, -40m to -65m.

The problem is the amount of water inflow to be expected during construction and the measures needed to deal with it. At low flows it may be sufficient to pump away and, when the lining is being concreted, to divert flows so that they do not affect the concrete.

At high flows a programme of probing ahead and grouting becomes necessary. In this rock there will be problems in finding the fissures to grout. (Compressed air working could not be used because a pressure of more than four atmospheres would be needed). In the small size of tunnels proposed, a machine, if used, would have little space free for probing ahead and grouting; if much of this was expected, the alternative 'drill and blast' method using the new-style hydraulically-operated drilling equipment would be able to maintain nearly as high an output, but this may cause greater water flow rate to occur.

For comparisons of progress one may look to recently completed schemes in rock, e.g.,

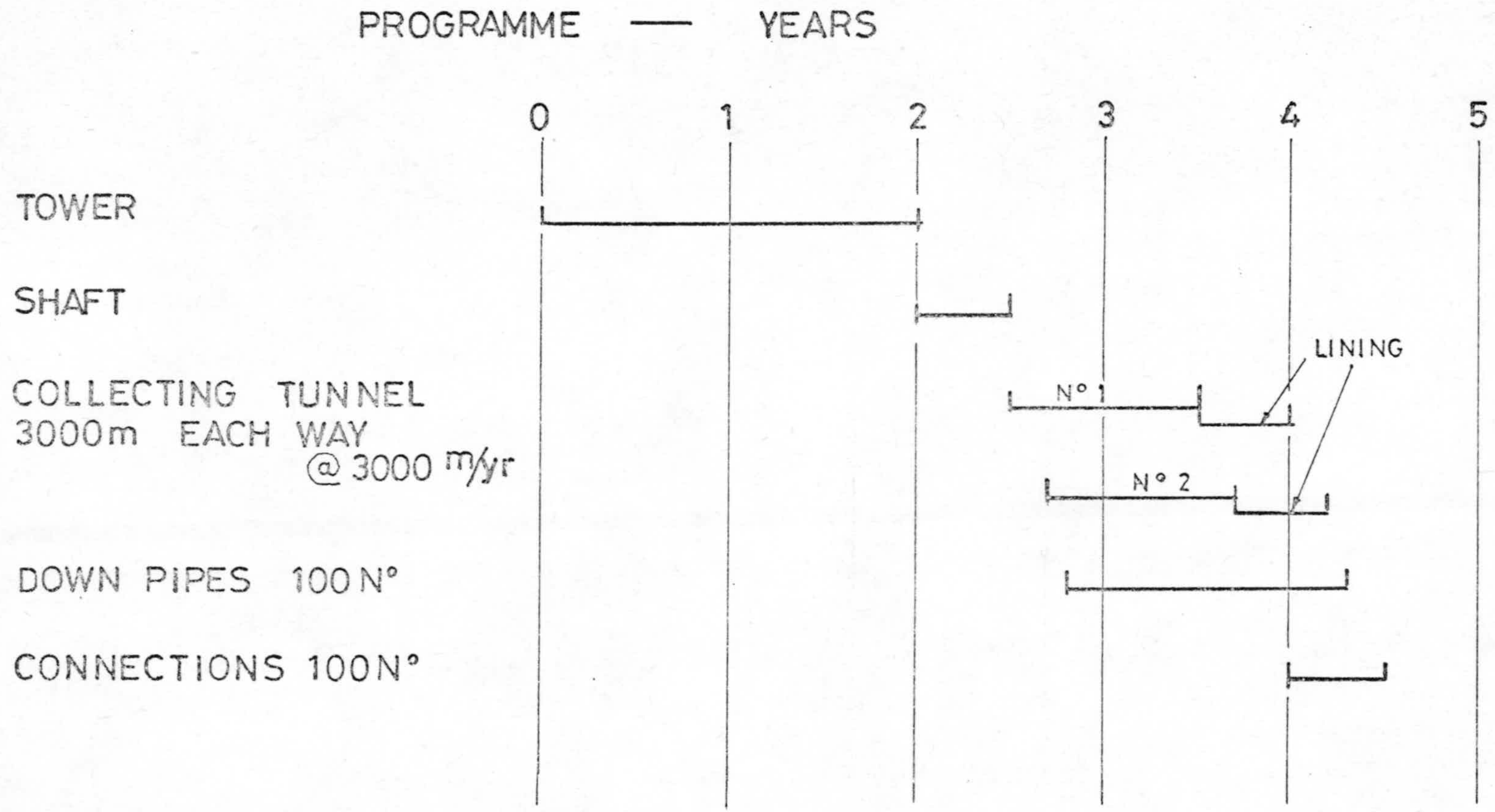
a) Kielder - drilling by machine, some roof support, the Robbins machine averaged 80m/wk but allowing for harder rock in this study take 70m/wk, 3.3m internal diameter, lining followed after excavation complete (Ref. 9).

b) Megget - overall average 62m/wk for excavation, driven by drill and blast, 3m wide by 3m high, very little roof support, lining followed later.

c) Severn-Wye - 3.67km in 30 months over 4 faces, averaging 12m/wk, includes dealing with inflows at the face mostly by grouting; because of slow advance, lining constructed in precast concrete as excavation advanced, part-face machines used, 12m/wk also achieved by drill and blast method.

For the Severn-Wye tunnel, by working more hours per week an output of 15m/wk could have been achieved. This might be taken to be the slow rate of progress in bad conditions for the current scheme. The alternative would appear to be to drill and grout from the seabed, but even in dry conditions the results may not be as good as grouting ahead from the tunnel although it would cause fewer interruptions to the tunnelling work. Furthermore the cost would be very high, particularly in these adverse sea conditions.
In considering this slow rate of advance one is led to suggest that an extra tower and shaft should be used for construction purposes in order to drive the access tunnel from three faces instead of one. This could also be achieved, though to a lesser extent, by placing the one tower at, say, 0.9km from the junction with the connecting tunnel.
For present purposes, therefore, a range from quite good to quite bad rock will be considered, i.e., rates of progress of 3000m and 750m per annum; in the latter case both one and two towers will be considered.
This gives 5 possible programmes for consideration and these are illustrated in Figs. 6.2 to 6.6 respectively.

2. The Down-pipes to the Tunnel
These will be put down to one side of the tunnel, lined with steel tube and topped by a valved dome. This part of the work is priced independently of the cylinders' anchors, though there is the possibility of a substantial saving if the piling jack-up platform (Section 4.2) carries out both sets of drilling as it progresses.
3. Cross-connection between Down-pipes and Tunnel
It is assumed here that, like the down-pipes, these are required every 60m though they would both be more widely spaced to suit wider gaps between the cylinders (Section 2.5), and it is also possible that the outputs of more than one cylinder may be fed to one down-pipe.

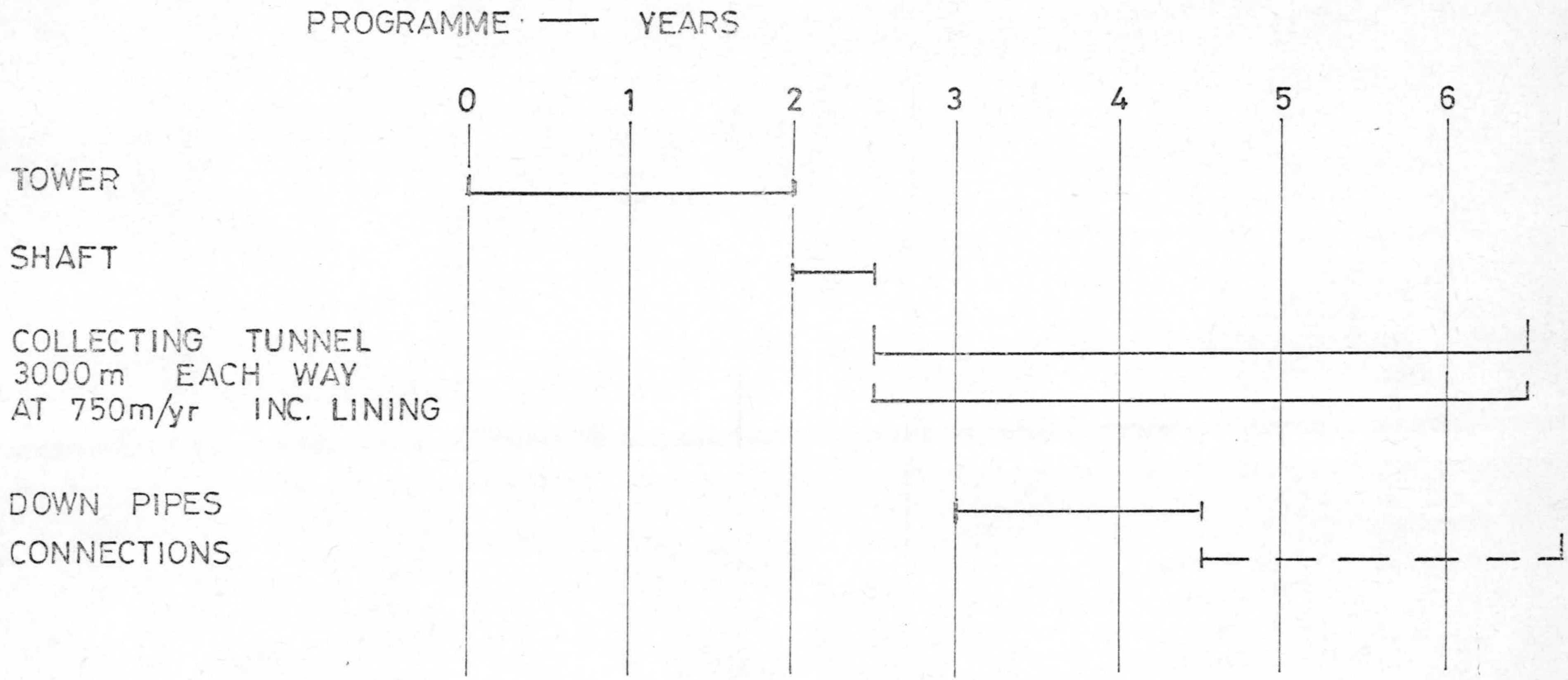


GENERATOR ON TOWER

IN GOOD GROUND 4 ½ yrs

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.2

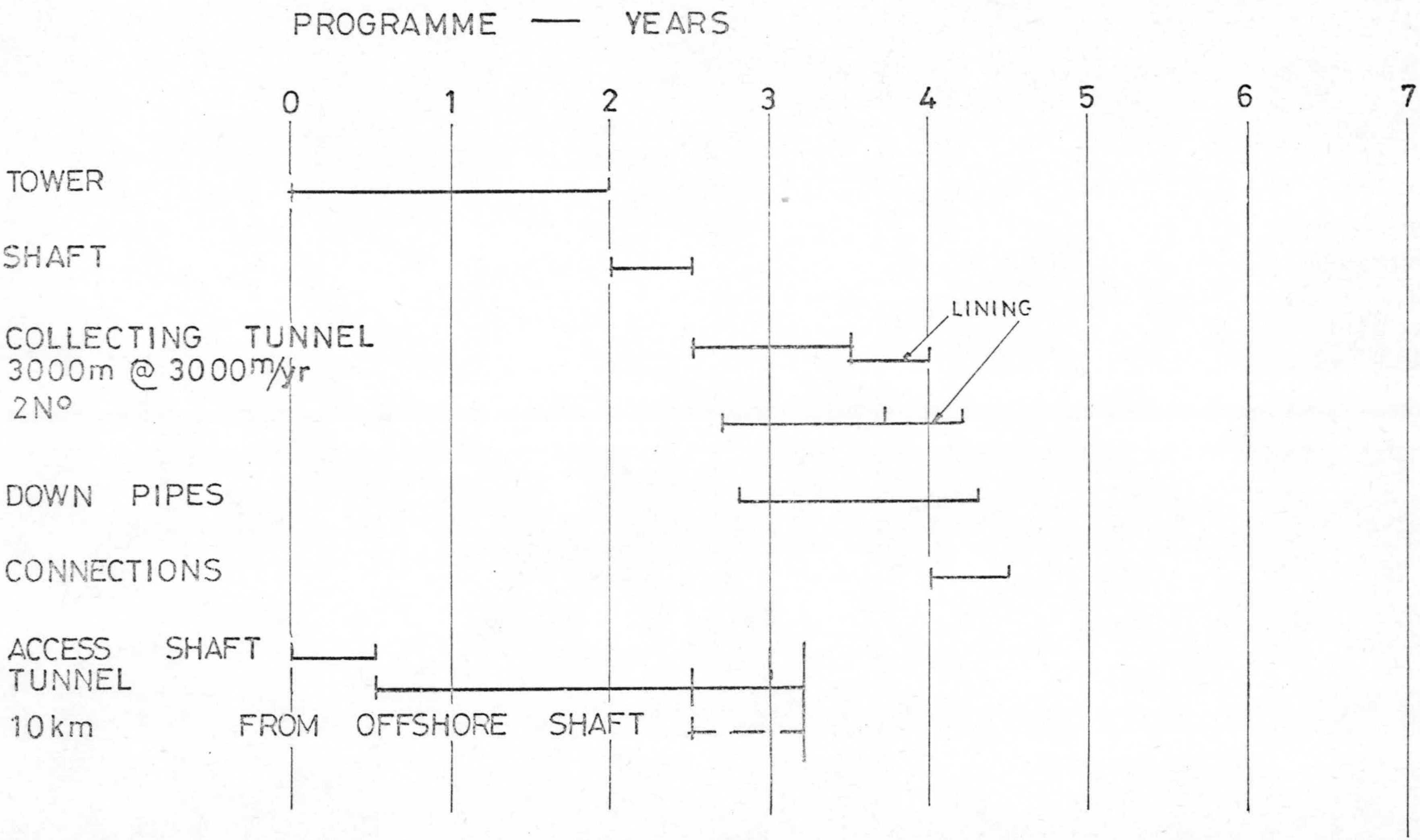


GENERATOR ON TOWER

IN BAD GROUND
TOWER @ 10km

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.3

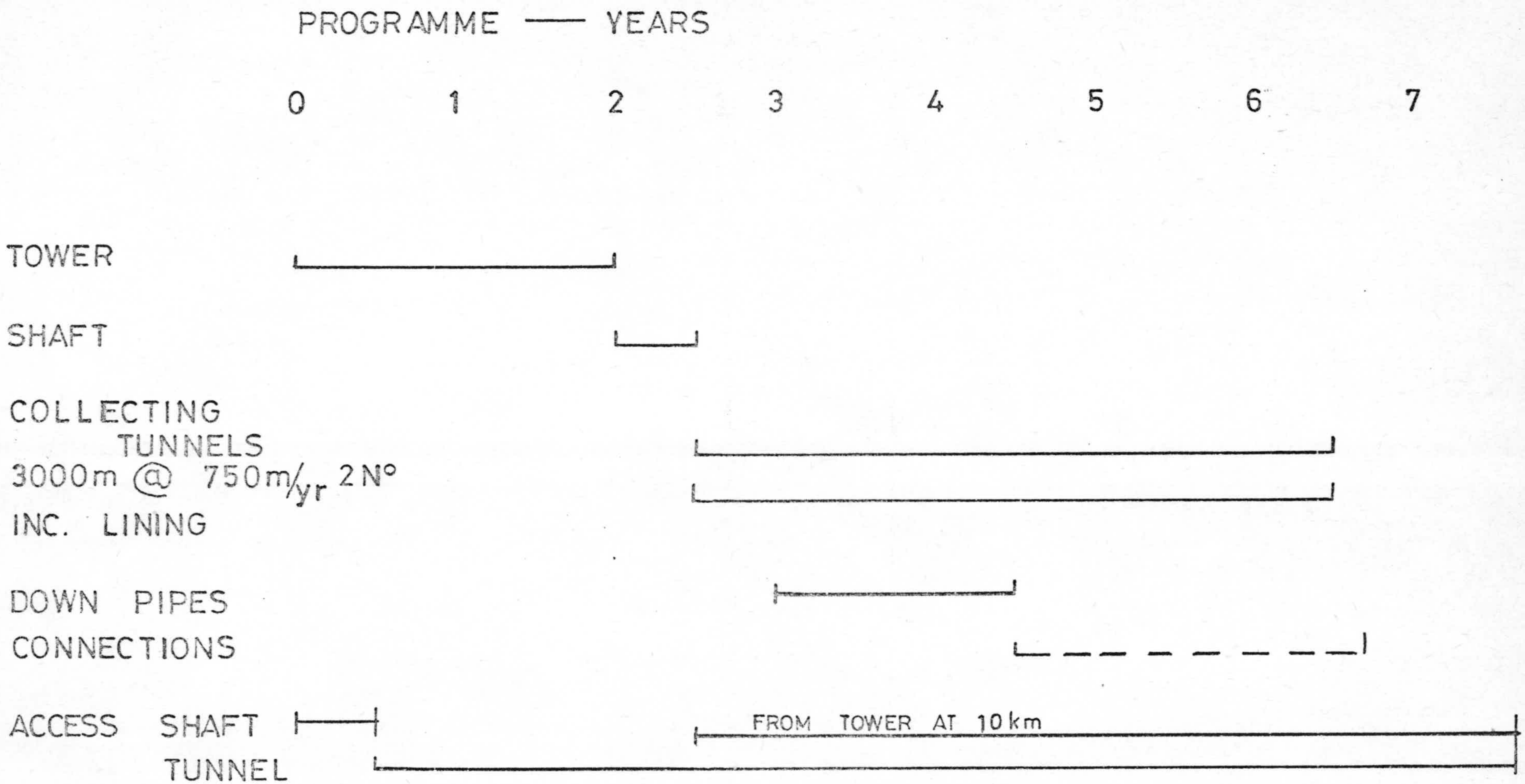


GENERATOR ON SHORE

IN GOOD GROUND

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.4

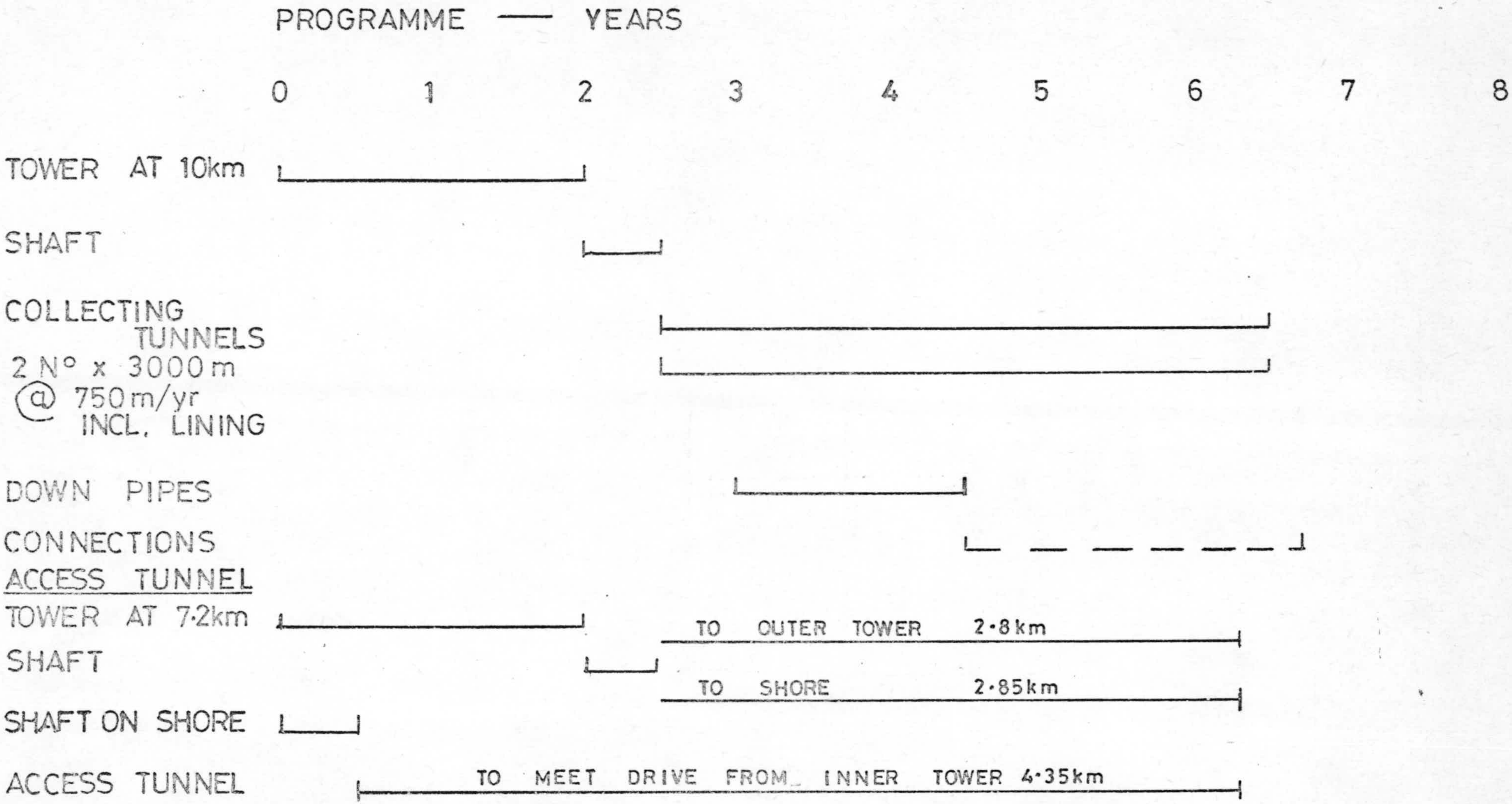


GENERATOR ON SHORE

IN BAD GROUND
TOWER AT 10km

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.5



GENERATOR ON SHORE

IN BAD GROUND
TOWERS AT 7·2km and 10km

BRISTOL SUBMERGED CYLINDER SCHEME

FIG 6.6

The cross-connections cannot be installed during the excavation cycle because of the interference this would cause with work in the small diameter tunnel, but it could be done during the lining period. In the case of the slow advance they may be included during the general advance. Four gangs would work on these connections in each tunnel for about six months.

Access Shaft

Assume that the Depth = 18m

Internal Diameter = 6m

External Diameter = 7m

This is the first place where the nature of the rock, offshore, will be tested. The use of compressed air is not suitable because of the high pressures required, hence grouting will be necessary.

Allow: Grouting	-	2 months
Excavation	-	2 months
Lining	-	1 month
		5 months
	say	6 months total

6.3c Costs

1. Collecting Tunnel

Average price of recent contracts	- £1.25M/km
Allow extra for heavily reinforced lining	- £0.65M/km
Total in good rock	£1.90M/km
Excavation portion of £1.25M/km is £0.75M/km	
If this is 4 times more costly add	- £2.25M/km
Total in bad rock then becomes	£4.15M/km

2. Downpipes - approx. 4 days each

100 No. = 67 weeks (6 days/week)	
Allow for bad weather conditions in Atlantic	
Hire charge on long term for jack-up platform	= £400/hr
times 144 hours x 52 weeks	= £3M/yr
This includes services of all tugs and support ships, helicopters, drill equipment, etc., plus drivers to operate equipment	
For 67 weeks, total cost	= £4M, say
Add £280,000 for domes and pipes, giving for 100 devices a total of	£4.28M

3. Cross-connections - allow 3 wks each with gang
@ £3000/gangweek
For 100 No. total = £0.9M

4. Shaft below Generating Tower = 18m deep by 6m dia.
@ £5000/m = £0.09M

5. Accommodation & Services of Generating Tower say £0.05M/wk

(This allows for the dedicated use of a helicopter and supply vessel and accommodation for 100 men)

for additional shaft and Generating Tower say £0.03M/wk

Summary of Costs - Scheme 1
(Collecting tunnel with cable to shore)

	£M Good Rock Tunnel	£M Bad Rock Tunnel
Progress in tunnel	60m/wk	15m/wk
Overall programme time	4.5 yrs	6.75 yrs
Programme Fig. No.	6.2	6.3
1. Collecting tunnel, inc. lining, 2.5m i.d.		
6km @ £1.90M/km - good rock	11.4	
@ £4.15M/km - bad rock		24.9
2. Downpipes - 100 No.		
Barge, support ships and men	4.00	4.00
Domes and pipes	0.28	0.28
3. Cross-connections - 100 No.	0.9	0.9
4. Shaft below Generating Tower	0.09	0.09
5. Cost of accommodation & servicing on Generating Tower		
Good rock - 2.5 yrs @ £0.05M/wk	6.5	
Bad rock - 4.75 yrs @ £0.05M/wk		12.35
Total cost of above to service 100 cylinder (0.2GW) installation	£M23.17	42.52

NOTE The cost of the tower has not been included as this will form the permanent platform for the turbo generators and was priced separately in our October '79 Report.

Summary of Costs - Scheme 2
(Collecting tunnel & Access tunnel to shore)

	£M Good Rock Tunnel	£M Bad Rock Tunnel 1 tower	2 towers
Progress in tunnel	3km/yr	0.75km/yr	
Overall programme time	4.5 yrs	7.75 yrs	6.75 yrs
Programme Fig. No.	6.4	6.5	6.6
1. Collecting tunnel, as before	11.4	24.9	24.9
2. Downpipes, as before	4.0	4.0	4.0
3. Cross-connections, as before	0.28	0.28	0.28
4. Shaft below tower, as before	0.09	0.09	0.09
5. Cost of accommodation, on Gen. Tower	6.5		
Programme 6.5, 5.75 yrs @ £0.05M/wk		14.95	
Programme 6.6, 4.75 yrs @ £0.05M/wk			12.35
6. Shaft on shore 50m @ £5000/m	0.25	0.25	0.25
7. 10km Access tunnel, 3.3m dia.			
Good rock £2.4M/km	24.0		
Bad rock £4.7M/km		47.0	47.0
8.* Additional tower for bad ground £25M, divided by 2 uses			12.5
9. Shaft at additional tower say 22m @ £5000/m			0.11
10. Cost of accommodation & servicing additional towers Programme 6.6, 4.5 yrs @ £0.03/wk			7.0
Total for 100 units (0.2GW)	£M46.52	91.47	108.48

* Only the additional tower costs are included as this structure is not required as part of the permanent works. See Note of Scheme 1.

6.3d Conclusions

From the above it will be seen that the possible costs of tunnels for hydraulic transmission vary considerably. If we compare the collecting tunnel only (Scheme 1) with the estimated cost of 1 metre bus mains quoted in the October 1979 Report we have :

Cost of pipe mains £1.085M per device

Cost of tunnel £0.232M to £0.425M per device

Even with very generous allowances for contingencies on this very preliminary tunnel estimate it is clear that the subject warrants further study in the next phase.

not comparing like with like ie Large bus mains

6.4 Marine Growths

We are conscious that marine growths on structures in the temperatures, depths and light conditions foreseen for wave energy devices in Scottish waters may, in the medium and long terms, act to the detriment of the performance of various components of the cylinder device. We are taking steps, in conjunction with the Scottish Marine Biological Association, to quantify these effects as far as known evidence at present permits, and we are identifying how further information could beneficially be collected. We are also taking steps to ensure that we have available alternative means of performing those functions of the device that marine growths appear, on present evidence, likely to threaten in unacceptably short periods. If further field information confirms these earlier suspicions we will then modify the design accordingly.

Our principal concern at present is for the efficiency and lifetime of the strainers, pumps, valves, bus mains and turbine of the hydraulic transmission system if this uses seawater as the working fluid. Present evidence strongly suggests that it will not be possible to prevent larvae of, for example, mussels from passing the strainers, and that these will survive in the main notwithstanding the fact that it is enclosed and operating at very high pressure. The velocity of flow will not deter mussels (which are bi-valve molluscs) and various crustaceans from clinging to and colonising first any sheltered areas like bends, valves and joints in the transmission system, then growing into communities sufficient to influence the hydraulic efficiency of the system.

Although it is not yet clear how quickly any material influence to the efficiency will occur, it has been provisionally estimated that within three years the cross-sectional area of the mains could be reduced to such an extent as to seriously prejudice the viability of this critical link in the whole power transmission system.

Because it will clearly not be possible to remove growths on this scale by physical means, we have sought evidence on the alternative of chemical treatment, especially chlorination. Opinions vary about the likely success of this, though it is acknowledged that dosing would have to be at least frequent if not continuous to ensure an adequately successful operation.

The problems of dosing on this scale and in this location have not been assessed though we do not regard the prospect of having to do this with much favour. The consequences for the turbine of receiving the hard detritus of marine organisms is uncertain and must be considered, though if growths within the system are prevented by dosing from the outset the strainers at the intakes to the pumps will adequately protect the turbines. (Water will flow from the furthermost pump to the turbine in a time of about 15 minutes, though when the system is being installed, or when component parts are being serviced or repaired, seawater could be static in the pipeline for periods of weeks).

Although it is too early to judge the magnitude of these problems of using seawater as the hydraulic transmission fluid, we are also considering freshwater and oil. These liquids would make it necessary to install return mains. The pressure in these would be small but there would be an energy loss which an open circuit system would avoid. Seawater could also be used in a closed circuit system, when it would presumably not yield the problems referred to above. Its particular attraction is its ready availability and zero cost, though the difficulties that it could bring with it seem certain to appreciably offset its basic advantages.

In addition to the above-mentioned problems of marine growths, which are not confined to the cylinder device but are common to all those for which a hydraulic link to a turbo-generator is needed, we are also exploring the possible ways in which they, together with the various species of Laminaria that may grow at the water depths (light conditions) that apply, may detrimentally colonise the outer surfaces of the power takeoff units. In Section 3.5, it was noted that such colonisation is unlikely to affect the performance of the mooring rodes, though it is not clear if the uses we may have for pins and bearings would also be unaffected. The shafts and glands of the pumps and springs would almost certainly have to be protected by gaiters, but we have yet to ascertain whether the performance of any unit in which heat is generated, and which requires to be cooled by sea water, would operate less successfully if its outer surfaces were covered in various ways. It is tempting to think that warmed surfaces may attract more prolific growths of many species, also some which might not otherwise take root. However, it may be the case that the surface operating temperatures of some components may be too high to permit growths to become established, or to survive if they start to form when these installations are not operating.

6.5 Choice of Turbine

Our reference design incorporated a Francis turbine driven by sea water at about 500m head. The possibility of using Pelton turbines operating at, say, 500m head is now being discussed with Sulzer Bros. (The highest head conventional hydro-electric installation currently in service is at Reisseck, Austria, where a Pelton turbine operates under 1766m). A 100 MW machine would run at about 500 rpm and have a diameter of 3.5m. The particular problems of erosion using sea water in these conditions are being evaluated.

On the information at present available to us there does not appear to be any particular advantage in using a multi-stage Francis machine in preference to a standard Pelton design. The highest multi-stage Francis installation is thought to be 672m at Roeshag in Austria, which is only 38% of that of the Reisseck station. One disadvantage of the Pelton system is that the impeller operates at atmospheric pressure which means that, in the present application, a proportion of the available pressure head will be lost according to the height at which the turbine nozzles are located above sea level. In a closed cycle hydraulic transmission system (Section 6.4) the liquid must either be pumped or sucked back to the pumps of the power takeoff unit. Alternatively it could be collected in a reservoir located sufficiently above sea level for gravity to provide at least some of the motive force, but any advantage that it would seem at first sight that this arrangement might offer would be offset by the extra elevation required for the turbine to be above this level. The Francis turbine is of a type that would allow the discharged liquid to be piped directly back to the pumps, the only precaution being that the pressure upstream of the pumps should at no point be low enough to allow vapour to come out of solution.

6.6 Electrical Transmission

We have carefully considered the Reports that have been issued on possible electric power transmission systems linking offshore wave energy devices to a distant network connection. While we do not feel well qualified to comment on the detailed issues involved, we intend to take advice on the fundamental issues that appear to make appreciable differences to the unit costs that, putting it simply, must be added to the cost of electric units at the turbine to give the equivalent figure at the network. We are concerned that transmission costs well in excess of 1p/unit are quoted which is a severe penalty for an intermittent source to have to bear, at least by present costs.

It appears that this figure is high for two reasons, first because of the high cost of underwater cable, and second because wave energy devices operate at low load factors of about 35%.

In Section 6.3 we reported our first estimates of tunnelling costs, and in Section 6.3d show that a tunnel may be a more economic solution for power transmission than a pipeline. We have yet to make the equivalent comparison with a cable, but this subject together with the choice of optimum power takeoff rating, hence load factor, will be considered in detail in Phase 4.

SECTION 7

LIFETIME OF INSTALLATION

In our Report dated October 1979 we estimated that the life of a 2GW installation might be 20 years but that some components like the rodes would be replaced twice during this period. WESC's Consultants, Rendel, Palmer and Tritton, subsequently suggested we should work on a 25 year lifetime, and we shall be looking closely at this suggestion as the preferred components for the device and the conditions under which they have to operate are more precisely defined during Phase 4 of this study.

Various studies relating to the reliability and lifetime of the device have been reported in earlier Sections of the present Report insofar as work on them has proceeded, for example on fatigue analysis in Section 3.1, on the durability of metals in sea water (Section 5.5), and on marine growths over and within important component parts of the device (Section 6.4).

The problems involved are clearly considerable and we cannot pretend to have yet made the comprehensive studies that these topics demand before sound estimates about the real reliability and probable maintenance requirements of the device can be judged. This is rightly a part of Phase 4 of this study, and it will be dealt with in detail at that stage.

The fatigue life of reinforced concrete can be considerably increased by working to Det Norske Veritas' "Rules for Design, Construction and Inspection of Offshore Structures, Appendix D dealing with Concrete Structures", and the References mentioned therein. That study uses conventional S-log N methods, and for S (loads) we need to have a full representation of the wave heights, their associated periods and number of occurrences over, preferably, the design lifetime of the structure.

No further work has been done on maintenance during the period of this supplementary contract other than to study the recent Easams Report to WESC. There is much useful information in this report which will be of value when we carefully consider maintenance during the next phase of our work. It would be premature to carry out further work on maintenance until power takeoff methods have been reviewed, component durability assessed and connections properly designed. Similar remarks apply to reliability assessments and these are also in our ongoing programme.

SECTION 8

ENERGY ESTIMATES

8.1 Influences of Tuning Period and Submergence on Capture Efficiency

Fig. 2.5 of our October 1979 Report strongly suggests that small amplitude wave theory accurately predicts the performance of the cylinder in low waves across the range of frequencies relevant to sea waves. Figs. 7.10 and 7.11 of our earlier Report are based on the same theory and show how the diameter of the cylinder, its submergence below still water level, and its tuned period (held constant in each case) influence the shape of the efficiency characteristic across the same range of frequencies. Fig. 8.1 of the present Report presents equivalent data for an 11 second tuning period, this period having been chosen to see whether a value higher than 8.0 and 9.8 seconds used in our October 1979 Report might give a better match to the recorded annual S. Uist wave spectrum (Fig. 7.8 of that Report clearly indicates that tuning at 8s is too low for maximum efficiency).

The complete answer to this very important question will not be known until capture efficiences in a representative range of real waves are measured during the forthcoming series of tank tests. However, some first indications of the trends to be expected have been produced by reinterpreting the data in Fig. 7.8 of our earlier Report for a 12m cylinder, submerged by 3m and tuned to 8 seconds, to other submergences and tuned periods. The theoretical envelopes referred to above have been used as guides for this purpose. Table 8.1 and Fig. 8.2 show the results obtained for average annual capture efficiency based on the 1976/77 scatter diagram as presented in Fig. 7.5 of our October 1979 Report. It is assumed that lateral energy capture is equally evident in each case though the magnitude of this was not measured in the tests in the Edinburgh wide tank from which Figs. 2.5, 7.8 and 7.9 of our earlier Report were obtained.

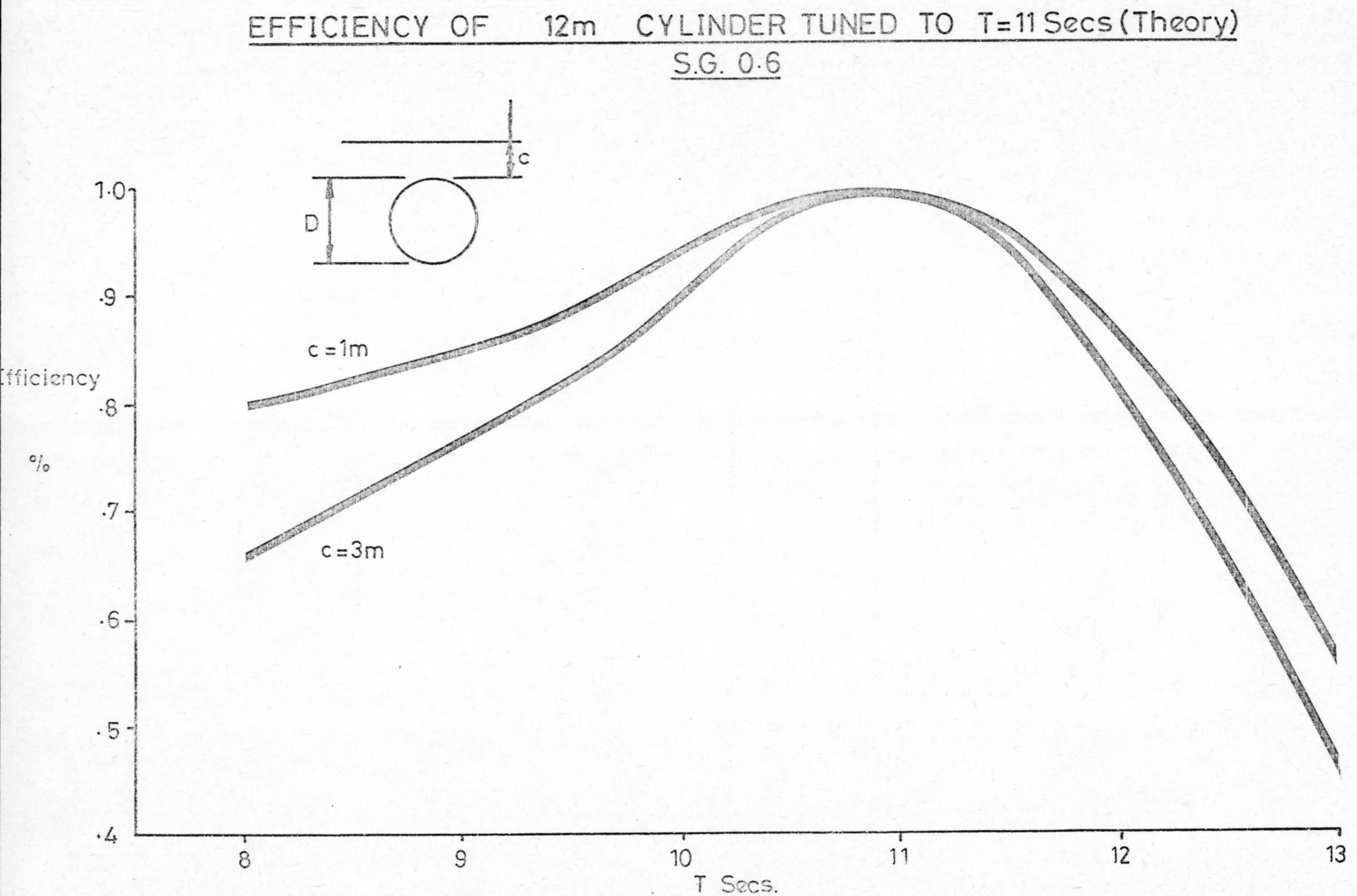
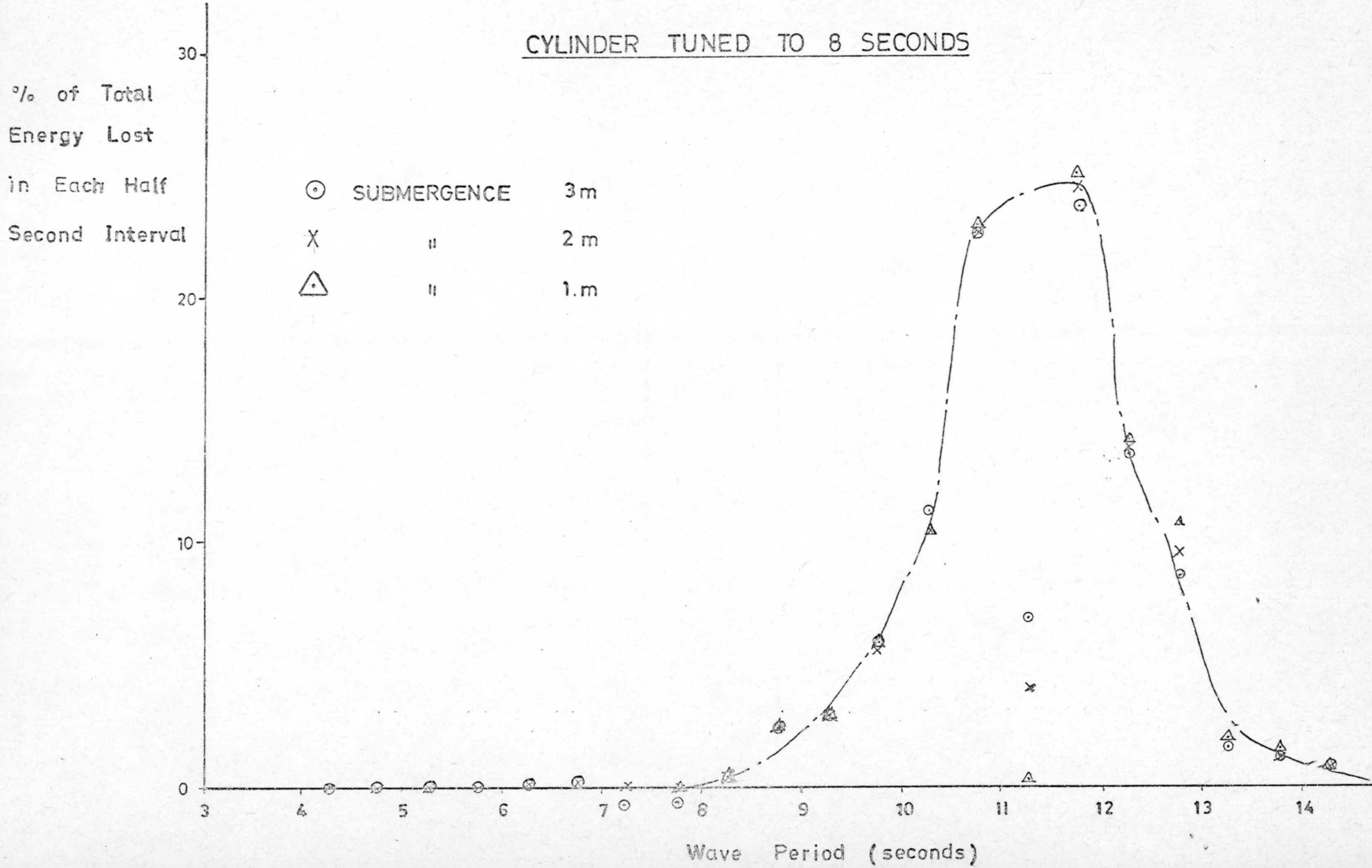

EFFICIENCY OF 12m CYLINDER TUNED TO T=11 Secs (Theory)
S.G. 0·6
c
D
1·0
·9
·8
·7
·6
·5
·4
Efficiency
%
c=1m
c=3m
8
9
10
11
12
13
T Secs.

FIG 8.1



CYLINDER TUNED TO 8 SECONDS
% of Total Energy Lost in Each Half Second Interval
SUBMERGENCE 3m
" 2 m
" 1.m
30
20
10
0
3
4
5
6
7
8
9
10
11
12
13
14
Wave Period (seconds)

FIG 8.2

Table 8.1 Average Annual Capture Efficiencies

Submergence (metres)	Tuning Periods (seconds)		
	8.0	9.8	11.0
1	53.2	63.4	64.2
2	49.8	60.9	Not calculated
3	47.0	58.0	59.4

These results suggest that there is little to be gained by increasing the tuned period from 10 to 11 seconds, though an intermediate period may give the optimum solution. There is a clear advantage in minimising submergence at any tuned period though it has yet to be determined by experiment whether, in severe waves, the maximum forces imposed on the device are notably sensitive to submergence. (Section 2.1 describes the motion of the cylinder relative to the phase of the wave. There is no particular reason to suspect that the cylinder is any more exposed to breaking wave effects when the submergence is only 1m than when it is more).

Our present view is that the absolute mean level of the cylinder should be chosen and kept constant, irrespective of the state of the tide. The minimum acceptable submergence (say 1m) will therefore correspond to at least the lowest astronomic tide, and lower if significant negative surges are regarded as a risk. In a location having a 4m spring tide range, which is typical of the Hebrides, the efficiency data given in Table 8.1 for a submergence of 3m would therefore correspond closely with the average conditions that, on present evidence, may be expected. The further tank tests planned for Phase 4 will look in detail at the effects of submergence and tuning period on capture efficiencies across representative wave spectra.

Figs. 8.2-8.4 show where energy is not captured by each combination of submergence and tuning period presented in Table 8.1 The data are given in terms of the proportion of all the energy that is not captured by each

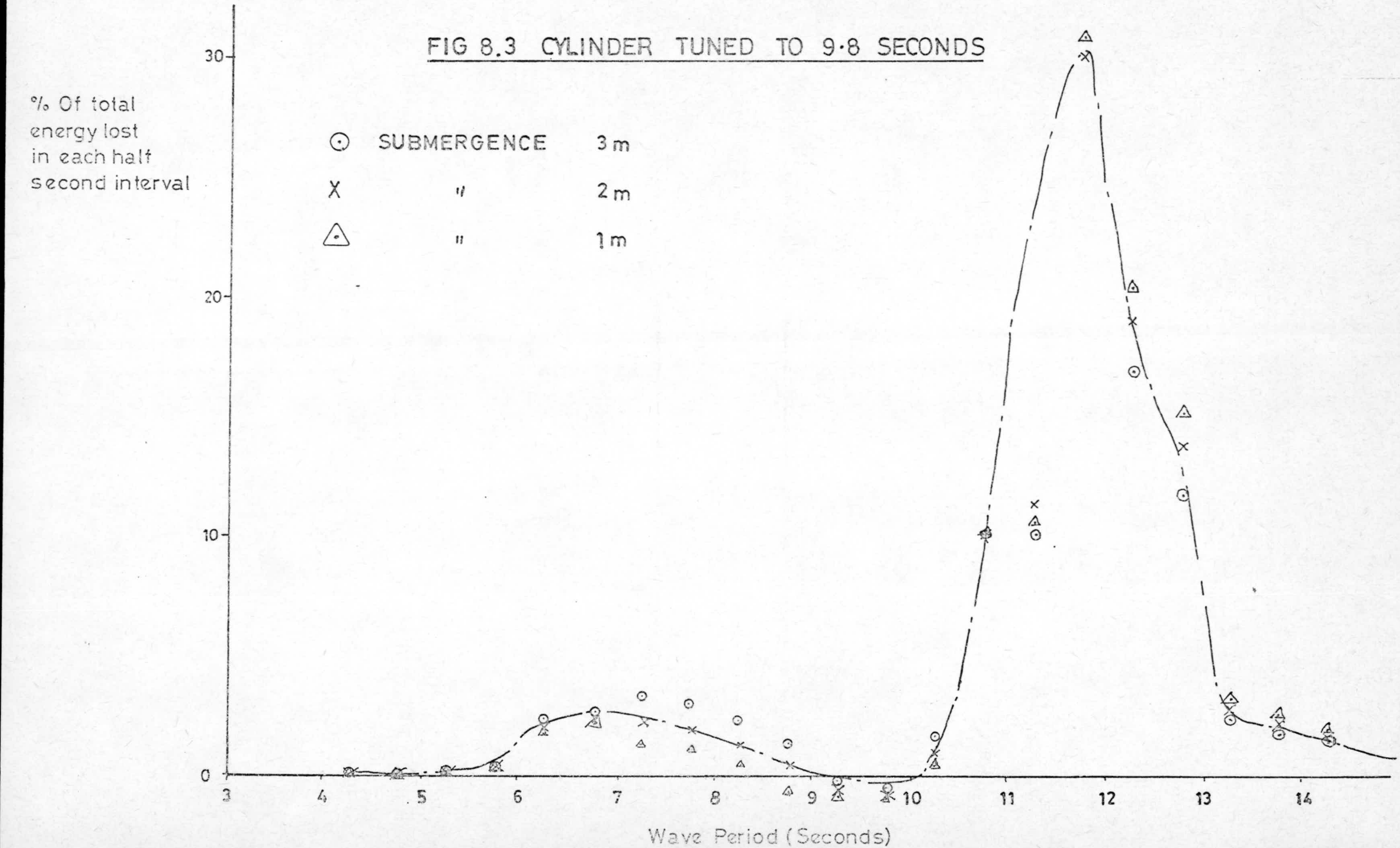


FIG 8.3 CYLINDER TUNED TO 9·8 SECONDS
% Of total energy lost in each half second interval
SUBMERGENCE 3 m
" 2 m
" 1 m
30
20
10
0
3
4
5
6
7
8
9
10
11
12
13
14
Wave Period (Seconds)

FIG 8.3

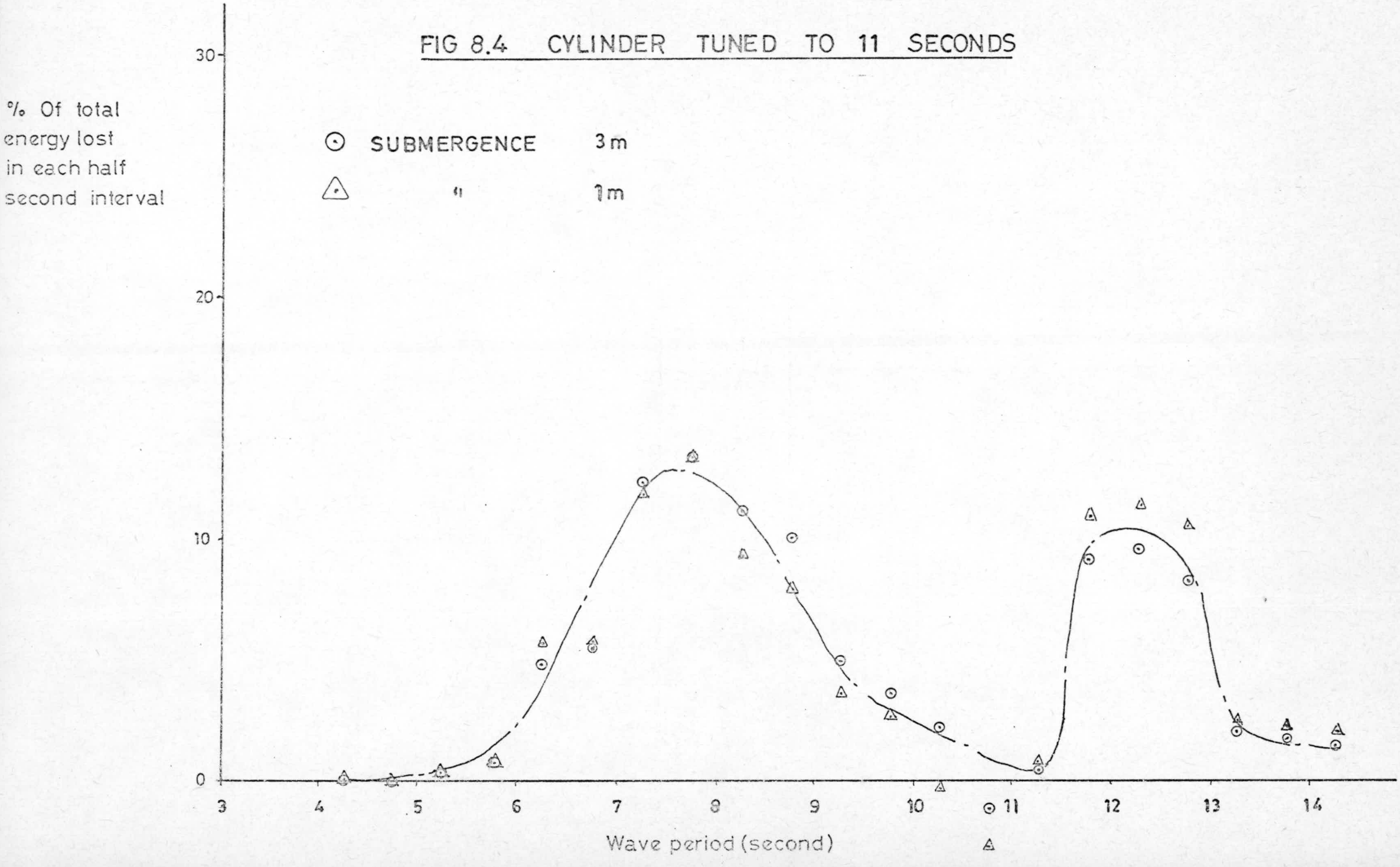


FIG 8.4 CYLINDER TUNED TO 11 SECONDS
% Of total energy lost in each half second interval
SUBMERGENCE 3m
" 1m
30
20
10
0
3
4
5
6
7
8
9
10
11
12
13
14
Wave period (second)

FIG 8.4

combination that occurs in each half second interval of wave period. The apparent discontinuity in all data in the wave period interval 11.0–11.5 secs is a consequence of the 1976/77 S. Uist reference wave scatter diagram (Fig. 7.5 of our October 1979 Report) containing relatively little energy in this interval : in particular, the proportion of high waves in this interval is small, hence the energy lost is low.

It would appear from Figs. 8.2–8.4 that submergence does not significantly influence the distribution of energy loss at any tuned period. An average line for each period has therefore been drawn and scaled to correspond with the actual energy lost at each tuned period. Fig. 8.5 shows the results, from which it is clear how the effect of changing the tuned period moves the distribution of energy losses within the band of incident wave periods. The more symmetric form of the line in Fig. 8.5 for the 11 sec cylinder might suggest that it would be more responsive to the introduction of variable tuning mechanisms, such as were mentioned in Section 5.4.

8.2 Effect of Cylinder Buoyancy on Capture Efficiency

For various reasons we have hitherto adopted a specific gravity of 0.6 for the cylinder. Principal among these reasons were :

a) that the efficiency spectrum is broadened by increasing buoyancy (Fig. 8.6);
b) that the average tension then introduced to the cables would be sufficient to prevent them going slack in all but the more severe waves.

Reference was made in Section 3.2 of the present Report to the possibility of using steel or composite rodes, either with or without articulation. If a rigid form was adopted, or if the cylinders were mounted off A-frames, the case for positive buoyancy to prevent slack would be removed. However, as Figs. 8.5 and 8.6 imply, the improved bandwidth associated with positive buoyancy would mean that, for example, a neutrally buoyant cylinder would fail to capture as much energy, especially at wave periods well away from the tuned period adopted for the cylinder. However, since the efficiency

DISTRIBUTION OF RELATIVE ENERGY LOSSES

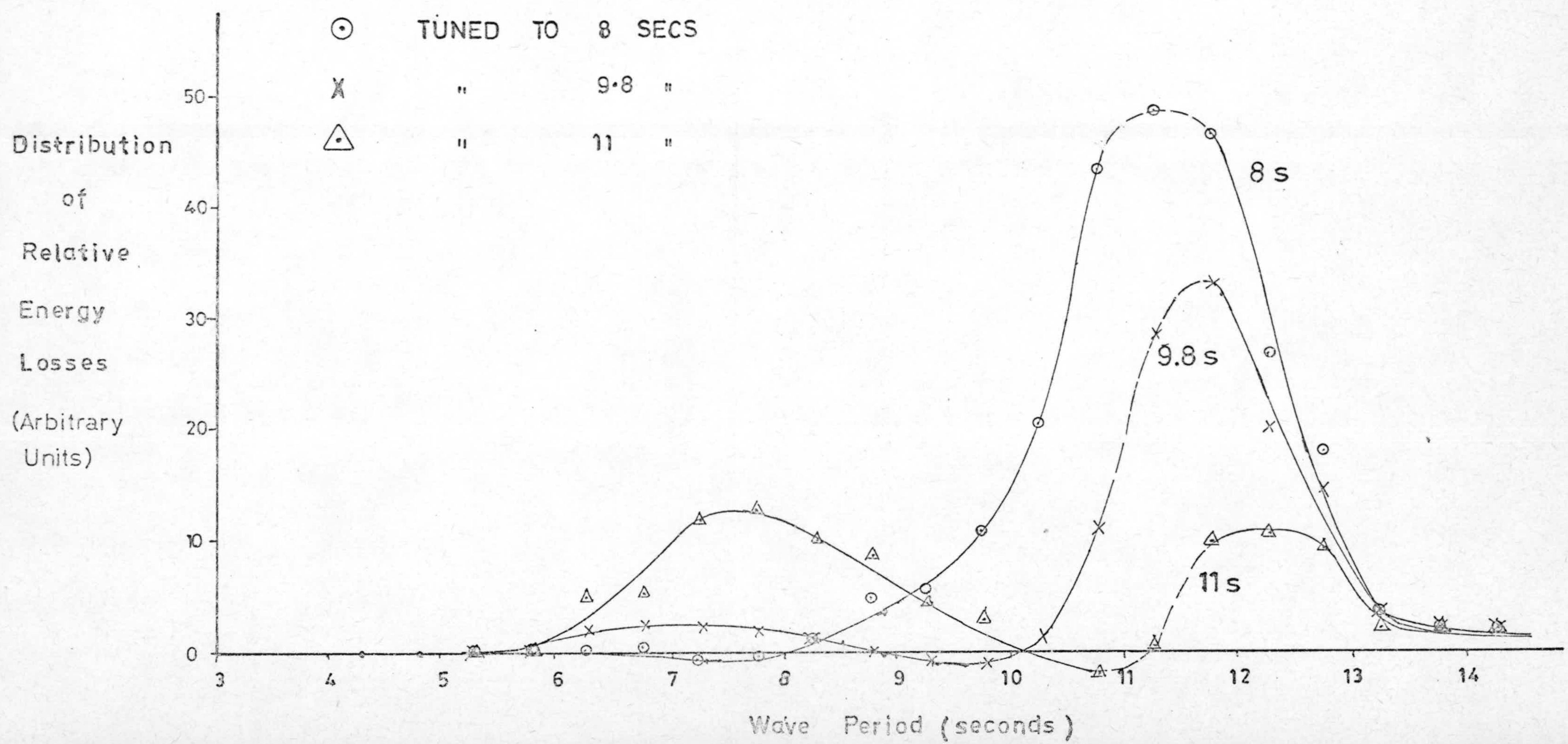
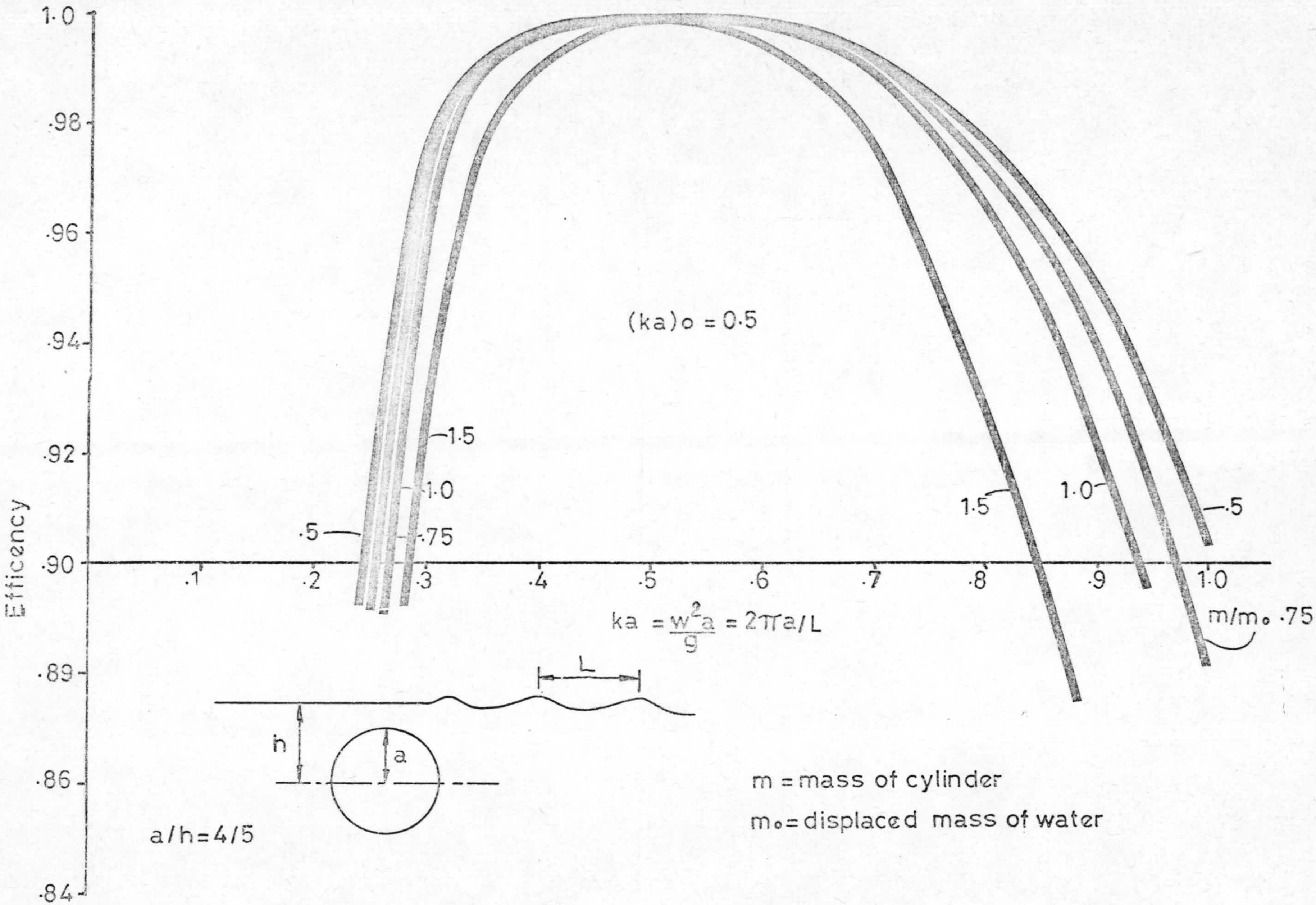


FIG 8.5



Effect on Efficiency of Varying m/m_o = S.G. of Cylinder

FIG 8.6

scale of Fig. 8.6 is exaggerated, this loss will be small. It is unlikely to be a significant reason for not considering how the structure for the device could be designed to suit a neutrally buoyant cylinder. Further attention to this option will therefore be given in Phase 4.

8.3 Cylinder Capture Efficiency in Directional Waves

Further to the statements in our October 1979 Report we have given more thought to this important subject. The analytical treatment given below will in due course be compared with the results for single and multiple cylinder efficiences measured in oblique seas in the tank tests scheduled for Phase 4, and we hope it will form the basis of a valuable method by which we may extend data collected in the tank.

It has been shown (Ref. 10) that for a body oscillating in a single mode, and taking power out of the waves in this mode, that the maximum power absorbed is

$$P_{max}(\beta) = |\chi_s(\beta)|^2 / 8B \qquad \ldots\ldots(1)$$

where β is the angle of incidence of the waves to the normal to a line of devices, $\chi(\beta)$ is the (complex) amplitude of the exciting force on the fixed body in the subsequent direction of motion of the body, and B is the damping coefficient (dependent on frequency) which measures the power required to force the body to oscillate. (If the body velocity is U, the mean power needed is $\frac{1}{2}BU^2$).

For N difference modes of motion, each capable of absorbing power, the result is

$$P_{max}(\beta) = \frac{1}{8} \underline{\chi_s}^* \underline{B}^{-1} \underline{\chi_s}(\beta)$$

where $\underline{\chi_s}$ is an N-vector and B is an N x N matrix. If the modes correspond to different bodies then B is a non-diagonal matrix. If however, the modes correspond to a single body and the motion in one direction does not affect the forces in another, then B is diagonal. This is the case for a long submerged cylinder oscillating vertically and horizontally normal to its axis. We have N=2 and $B_{11} = B_{22}$ (ω). The same is true for any two modes of motion of the (long) cylinder in a plane normal to its axis. In this case it is also true that the exciting force in the two directions is the same in amplitude and 90° out of phase.

Thus for the submerged circular cylinder oscillating in two modes and absorbing power in those modes, expression (1) above needs to be doubled.

The reason for the high efficiency in normally incident waves is the relation which exists between $\chi_s(o)$ and B.

Thus $$|\chi_s(0)|^2 = 4P_w B \ell \qquad .....(2)$$

where P_w is the power per unit crest length. Note that

$$P_w = \frac{1}{16\omega}\rho g^2 H^2$$ where H=wave height. Also ℓ is the length of the device parallel to the wave crests.

Thus $$P_{max}(0) = P_w \ell$$

and $$\eta_{max} = \frac{P_{max}}{P_w \ell} = 1$$

For oblique waves incident upon long devices an expression like (2) above does not exist unless the idea of a generalised damping coefficient is introduced (Ref. 11). For 3-D devices

$$\int_0^{2\pi} |\chi_s(\theta)|^2 d\theta = 8 L P_w B$$

which, for axisymmetric devices, gives the classical result

$$\frac{P_{max}(\beta)}{P_w} = \frac{L}{2\pi} \qquad \text{for point absorbers.}$$

To assess the effect of direction on efficiency we write expression (1) as

$$\frac{P_{max}(\beta)}{P_{max}(0)} = \frac{|\chi_s(\beta)|^2}{|\chi_s(0)|^2}$$

a form which does not contain B.

Now the efficiency of absorption $\eta(\beta)$ in oblique waves is :

$$\eta(\beta) = \frac{P(\beta)}{P_w \, \ell \cos\beta}$$

So,

$$\frac{\eta_{max}(\beta)}{\eta_{max}(0)} = \frac{P_{max}(\beta)}{P_{max}(0)\cos\beta} = \frac{|\chi_s(\beta)|^2}{|\chi_s(0)|^2 \cos\beta}$$

Thus the variation of efficiency with β is seen to depend upon the exciting force on the fixed body due to different wave directions.

The variation of $\chi_s(\beta)$ with β for the submerged cylinder is not known. It could be found but that would require a considerable research programme. Failing that, we can find examples in the literature. Thus Bai (Ref. 12) has evaluated $\chi_s(\beta)$ from Fig.9 in his Paper for a rectangular section cylinder in the surface, as set out in Table 8.2.

TABLE 8.2

	β =	0	15	30	45	60	75	90
ka = 0.1	$f_x(\beta)$	14.8	14.6	13.0	10.1	7.0	3.5	0
	$\frac{f_x^2(\beta)}{f_x^2(0)}$	1	.973	.77	.466	.224	.056	0
	$\frac{\eta\ max(\beta)}{\eta\ max(0)}$	1	1	.89	.66	.45	.216	-
ka = 0.2	$f_x(\beta)$	31	29.8	27.1	22.2	16	8.1	0
	$\frac{f_x^2(\beta)}{f_x^2(0)}$	1	.924	.764	.513	.266	.068	0
	$\frac{\eta\ max(\beta)}{\eta\ max(0)}$	1	.96	.88	.726	.532	.315	-
ka = 0.4	$f_x(\beta)$	52.3	51.3	48.5	42.2	32.8	18.9	0
	$\frac{f_x^2(\beta)}{f_x^2(0)}$	1	.962	.86	.65	.39	.13	0
	$\frac{\eta\ max(\beta)}{\eta\ max(0)}$	1	1	1	.92	.781	.5	-
	$\cos\beta$	1	.966	.866	.707	.5	.259	0

The function $f_x(\beta) = C\chi_s(\beta)$ is the horizontal exciting force on a long rectangular cylinder as a function of incidence angle β. (C is a constant, and is not needed).

Fig. 8.7 displays the result graphically.

Comments on Results

Although these results are only pertinent to a long rectangular cylinder absorbing power in surge they are the first theoretical results for any device showing the variation of efficiency with wave direction. As far as the submerged cylinder is concerned they should, for the present at least, be regarded as qualitative guides only.

Bai also provides curves of vertical force. These maintain almost constant values for β up to 45° and suggest that the relative efficiency would exceed unity in these cases. In other words $|\chi_s(\beta) / \chi_s(o)|^2$ decreases more slowly than $\cos\beta$ as β increases over that range.

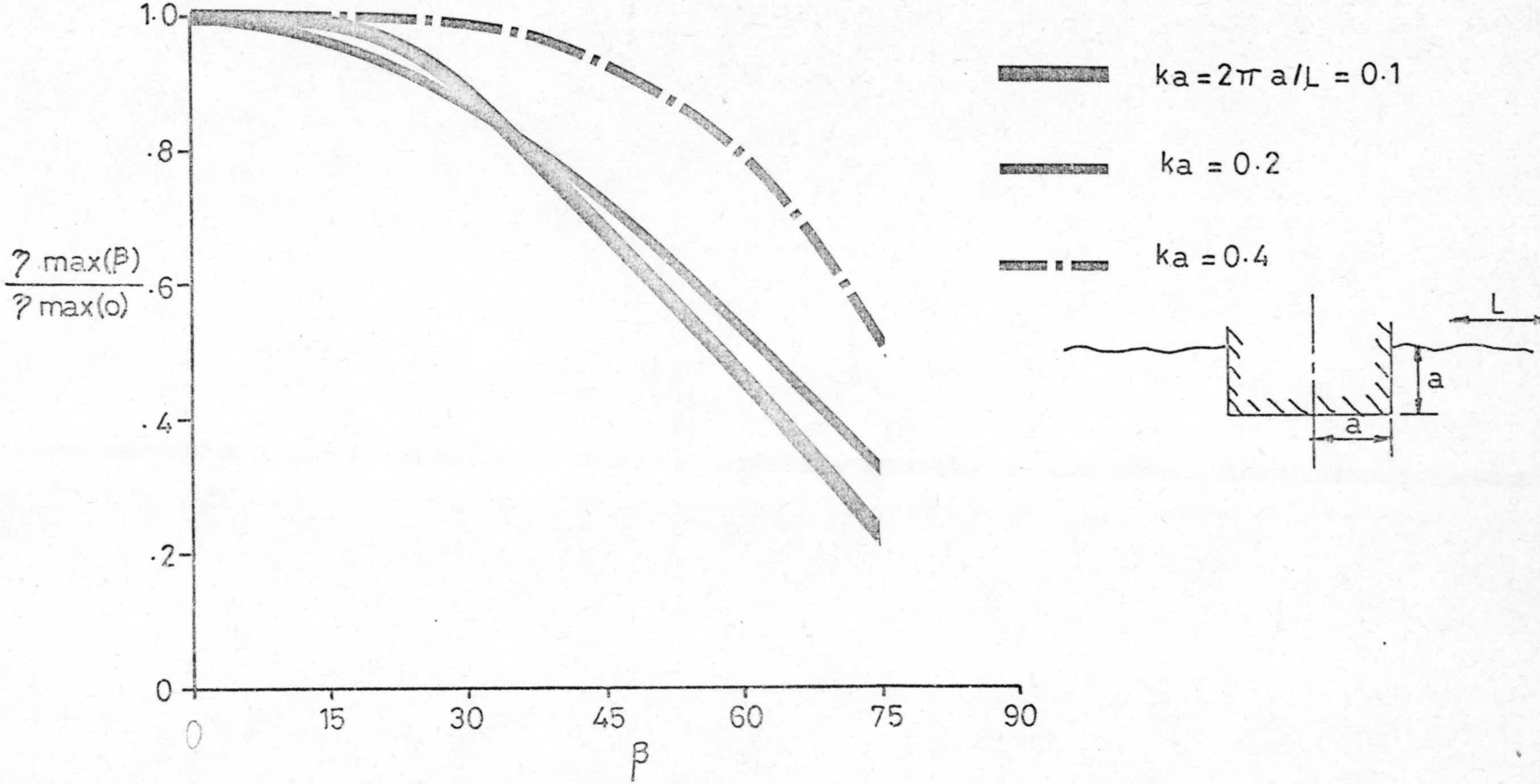


FIG 8.7

Variation of Relative Efficiency With Incident Angle.

This information may explain why the efficiency of the cylinder was observed, in the Edinburgh tank tests in May 1979, to remain at a high level even in oblique seas (Fig. 2.6, Oct '79 Report). The present work applies, of course, to many closely-spaced rectangular surface cylinders; it could be repeated for single submerged circular cylinders by using the results of Lepetit et al (Ref. 13) for that case, but comprehensive experimental data would still be needed to verify theory for the parameter values particular to the form and motion of the cylinder device, and hence to the wider use of that theory.

Our earlier and simpler attempt to anticipate the effects of wave direction on device performance were set out in Section 7.3 of our October 1979 Report. Following subsequent discussions with WESC's Consultants' some refinements to those concepts were made, as reported in the Consultants' recent report to WESC on our device. The theory presented above marks an attempt to improve our ability to predict performance in oblique seas according to, amongst other parameters, the size of the gaps between the cylinders. Although it must remain for the Phase 4 studies to demonstrate the optimum economic spacing of devices, this is, of course, related to overall annual performance in normal and oblique seas. (Another important hydrodynamic aspect of gaps was presented in Section 2.5).

8.4 Phase Locking

8.4a Theoretical Considerations

Most wave power devices are resonant systems which need to be tuned to the predominant wavelengths anticipated. Generally speaking tuning involves two parts :

a) the natural frequency of the device must be close to the wave frequency, and;
b) the (velocity proportional) damping constant of the pumps must equal the radiation damping coefficient of the device.

For small surface devices the first condition creates a problem. The required condition is

$$k = (m + m_a)\omega^2 \qquad .....(3)$$

where k is the spring constant, m is the mass of the device, m_a its added mass due to its motion in water, and $\omega/2\pi$ the wave frequency. Surface devices have a natural spring rate due to the change in buoyancy during a cycle. Thus a device having a waterplane area 'a' will, when submerged by an additional distance z experience an upthrust ρazg where ρ is the density of water. Thus $k = \rho ag$.

For example, a half-immersed sphere has $k = \pi\rho r^2 g$ where r is its radius, and $m = \frac{2}{3}\pi\rho r^3$ (if half submerged). Also, $\omega^2/g = 2\pi/L$ in deep water, where L = incident wavelength. Substituting in (3) gives

$$L/r = \frac{4}{3}\pi(1+\nu) \simeq 7.5$$

where $m_a = \nu m$, and ν is the dimensionless added mass for the sphere and can be taken as about 0.8 although it varies with ω.

It follows that to tune a sphere to L = 150m, a radius of about 20m would be required for <u>natural</u> tuning to occur.

Phase locking is an attempt to avoid this problem. Tuning is only a means to an end, the end in this case being to achieve maximum efficiency. This is achieved when the velocity of the body is in phase with the <u>exciting force</u> on the body. The exciting force is only part of the overall force, which is the sum of the exciting force (i.e., that on the body due to the incident waves <u>assuming the body is held fixed</u>) and the radiation force (i.e., that resulting from its own induced motion). Tuning properly achieves this coincidence of phase of exciting force and velocity.

For small surface bodies tuning is no good for the reasons argued above. Instead the body is artifically held in a wave crest at the top of its motion and released so that the phases are correct. This can be done using electromagnets in laboratory work and is the method used by Budal and Falnes in Norway. Similar techniques have been used at the C.E.G.B.'s Marchwood

Engineering Laboratories, and at the National Engineering Laboratory where a water column replaces the mass of the device. In all cases large increases in amplitudes of motion and resulting powers are achieved.

8.4b Drawbacks

1. It is not obvious how to separate the exciting force on the body from the overall force. For small bodies presumably the radiation force is small, but that is not the case for the cylinder device.

2. It is difficult to imagine a control mechanism enabling phase locking to take place in random seas.

3. The criterion for maximum efficiency of 'exciting force in phase with velocity' is based on assumptions of simple harmonic motions of given frequency. The technique employed in phase locking is to produce square wave motions which are not sinusoidal and thereby violate the very conditions under which the criterion that one is attempting to achieve is valid.

8.4c Is Phase Locking Appropriate for the Cylinder?

The cylinder, by being a submerged device, does not have a natural spring rate. We impose our own and, of course, choose it to satisfy the equivalent of expression 3 in our case. We already have a large structure. Phase locking enables small structures to be efficient by giving them large motions. Out of choice we would prefer not to have large motions, and we have now shown that in our case this is not essential for high efficiency (in contrast, limited stroke will almost certainly create problems for the smaller surface devices).

Phase locking is difficult enough in practice with one degree of freedom; with two it is even more so. Even if it could be implemented it would introduce snatch loads of a magnitude that could be totally unacceptable.

8.4d Non-linear Springs

The non-linear characteristics of the accumulator type spring (Section 5.6, Oct. '79 Report) have yet to be explored. It may be possible to create a variable spring load which would ensure that the device remains tuned over a wider range of frequency rather than nominally to just one (but see Section 8.1 for a discussion of the relevance of band widths). Extensive 2-D experiments will be needed to study this. Tests of efficiency for various electronically produced non-linear springs which could be achieved by the accumulator system should be carried out to find the 'best' non-linear pump characteristics. This subject will be followed up during the Phase 4 experimental programme.

8.5 Optimum Water Depth for Cylinder Device

The 42m water depth chosen for the offshore S. Uist wave recording buoy has hitherto been regarded as an acceptable location for the reference design of the cylinder device, firstly because real wave data are then available to give energy productivity estimates, and secondly because we have been able to accommodate the mooring and power takeoff system for a 12m submerged cylinder in that depth.

However, 42m is not necessarily the optimum depth, though for lack either of good data at other depths or the means of efficiently transposing the '42m' data to other locations it is tempting not to look to other depths. We are anxious not to be bound by these restraints because we can see operational advantages in moving to a depth of at least 50-60m. In the first place the wave climate is more energetic : WESC's Consultants suggest that up to 10% more energy may be available at the deeper end of this range. Furthermore the sea bed is thought to be more flat than at 40m, and wave action on the sea floor is less though currents may be stronger (Sections 2.3d and 2.3e).

The larger wave forces that would then have to be carried by longer rodes and the additional length of tunnel or seabed cable to shore needed (Section 6.3) may be a small extra price to pay for these additional benefits, especially since the device may in any case have to be designed for (local) depths of over 50m in order that an array is located in an average depth of 42m (Section 4.1).

The whole question of available wave climate slightly further offshore will have to be given much closer consideration during Phase 4 than has hitherto been possible because, for the various reasons mentioned above, the choice of water depth is likely to influence appreciably the economics of the device.

8.6 Standardised Device Parameters

The tendency for similarity criteria in mechanics to give rise to non-dimensional parameters like specific speed (for rotodynamic machines) and Reynolds number suggests that standardised numbers might also be produced for wave energy devices to give at least a first indication of their relative acceptability.

We have therefore considered a range of possible parameters based on quantities such as water volume displaced (both actual and virtual) and the weight of materials employed, but as yet have not come upon a more informative overall criterion than p/kWh. We suggest that this may be because the search is for a parameter which describes the total performance of the whole device, including all its mechanically very different functions and its capital and maintenance costs, rather than one function for which the governing criteria are in the main already well established.

SECTION 9

CONSTRUCTION METHODS AND COST ESTIMATES

The Maidenhead Workshop was rightly concerned that the wavepower programme should take full advantage of the unique opportunity afforded to the construction and manufacturing industries for mass production.

In the 1979 phases of the cylinder study we contented ourselves with sketching our methods of fabrication and possible layouts of construction facilities. More detailed study will be made of this during the ongoing programme together with the effect on cost estimates.

Our ongoing work on costs is indicating that the ratio of fabrication to installation costs could well reduce, there being more scope in the former for savings as indicated above.

We are in agreement with the generally held view that quantity production of the large number of relatively simple components required by the scale of installation of wave energy devices (about 1000 No. "2MW" units for a 2GW station) would lead to much lower unit costs than would be needed for a prototype installation of, say, 5 No. full scale units.

SECTION 10

CONCLUSIONS

The work in this phase of the study has been part of an ongoing programme and it is therefore not appropriate to seek absolute conclusions at this time. The additional information collected during the period covered by this Report has not changed our optimism or conclusions about the device as set out in our earlier Reports, though the information now available to us has shown where emphasis must be placed in future studies and the data needed to complete these satisfactorily.

A large proportion of Phase 3 has been devoted to theoretical and practical (Appendix B) preparation for the next series of tests in the wide tank. We have deemed it important to extend our knowledge of the theoretical performance of the cylinder when operating in its fully loaded conditions and to derive from that theory the forces that all components in the device will then bear. Thus, we shall be able to run more meaningful experiments in the wide tank and so collect data to refine the theory. This iterative process will provide the basis for our comprehensive appraisal of the device.

We are also very conscious of the urgency that must be given by the wave energy community to specifying design data on the following topics :

1. Wave steepness. All the information known to us shows that the limiting steepness of waves with specified probability of return decreases with increasing wavelength. It is essential to identify an acceptable relationship between steepness and wavelength so that individual device components may be logically designed according to the conditions in which they will be most susceptible to damage.

2. Fatigue spectra. The wave loading conditions to be experienced over the design life of the device must be specified. This information is vital for the selection and sizing of many components.

3. Currents. A likely upper bound for the strength of currents should be set in advance of the collection of detailed information. Allowance should be made for the possibility that locally strong currents will occur; the option of not siting devices in such places should remain open.

4. Marine growths. We have seriously considered the design implications for our device if the risk and consequences of growths turn out to be as serious as we have been given to believe. This subject clearly demands urgent enquiry and decision with concerted short-term investigation if necessary.

5. Rock conditions. Our enquiries continue to confirm the merits of rock anchors, and we are impressed by the apparent economic and operational advantages of tunnels compared with sea bed pipelines for hydraulic power transmission from individual devices to centralised turbines. Further information about rock conditions will in due course be needed to consolidate decisions about both of these expensive items.

Work on the construction and instrumentation of the models for the wave tank testing programme has progressed to schedule during Phase 3, for experiments to start in May this year (Appendix B). Information basic to the design and testing of a range of possible pumps and springs has also been assembled in anticipation of the needs of the total Phase 4 programme of which this is a vital element.

We are therefore confident that during the period covered by Phase 4 of the work, up to the end of 1981, we shall complete our theoretical and experimental analyses of cylinder behaviour, energy conversion and transmission leading to an optimised design so that, if called upon, we shall then be well placed to carry out a full engineering design in 1982 leading to prototype construction.

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APPENDIX A

WAVE DATA

We have made various attempts to clarify those aspects of real wave motion that we believe are important to the full appraisal of the cylinder device.

We are particularly anxious to know more about the steepness (Sections 2.3a and 10) and three-dimensional form of real waves. This information is mainly relevant to the performance and safety of the device insofar as it applies to long (rather than to short) waves.

For lack of experimental wide tank results we are not yet in a position to specify the threshold between these bands as far as our submerged structure device is concerned but it is unlikely to lie below 50m wavelength (T = 5.6 secs) and it may be over 100m (T = 8 secs).

A principal issue is the coherence of wave height along the length of the cylinder and how this reflects in its capture efficiency and the forces it has to carry. Ewing (Ref. 14) has presented some theoretical estimates of the average wavelength in a direction normal to the mean wave direction as a function of the average wavelength parallel to the mean wave direction, for various degrees of short-crestedness from an isotropic spectrum to swell-like conditions. Unfortunately, the only sea state between these limits quoted by Ewing is that of 'active wave generation...... near the peak of the spectrum', for which the transverse wavelength in waves of length 100m is about 250m. According to Ewing in these conditions an isotropic spectrum would have a transverse wavelength of 100m, whereas in swell-like conditions it would be 400m.

Using the value 250m, which is also the average of the limiting values quoted above, the crest (and trough) will be reasonably coherent for a length of 30-40m, whereas over 100m or more there will be a major change in the phase and amplitude of the motion. This suggests that from the point of view of this parameter, there is an upper limit to the length of the cylinder device. This value will not be clear until the device has been tested over a typical range of sea conditions (Appendix B), but from Ewing's work, bearing in mind that the performance of the device is best matched to longer rather than to shorter waves (Section 8.1) a cylinder length of about 50m appears justified.

Our theoretical work on the directional capture efficiency of the cylinder (and other) devices is continuing (Ref. Section 8.3) and it is hoped that a directional efficiency spectrum will soon be available for a given number of cylinders, based on a simplified theory. This will enable the energy captured in oblique and mixed seas to be estimated given, for example, the predictions made by the Meteorological Office for incident energy density over a range of frequencies and directions at a site west of S. Uist. It should be possible to use this technique to explore many very important characteristics of a wave energy station, for example, how the energy captured will vary from hour to hour and from day to day in different climatic conditions. This will allow the typical monthly and seasonal characteristics of the station to be estimated, hence the economic worth of the electricity produced determined according to its likely power levels, durations and timings.

In Section 8.5 we gave reasons why the cylinder might better be sited in a depth of about 60m rather than 42m. An additional advantage may be that a smaller though still unknown percentage of the local wave motion is a consequence of backscatter from the somewhat rough seafloor landwards of these locations. A significant component of backscatter would detrimentally affect the performance of the device, though the extent of this influence and the magnitude of backscatter present in different sea conditions and at different depths is not clear.

APPENDIX B

PREPARATION FOR 1980/81 WAVE TANK TESTS

B.1 Content of Test Programmes

Now that the general form of the device has been defined it is both possible and necessary to carry out a comprehensive tank testing programme to study its performance over a wide range of representative wave conditions and device parameter values. The programme scheduled for 1980/81 includes two parts and has required manufacture of two types of cylinder, namely an 'active' unit whose behaviour may be controlled and a physically identical 'passive' unit which has only simple mechanical spring and damping systems. Figs. B.1a and B.1b show the arrangement of these units for the two test programmes. The units needed for the first part of the part of the programme are now nearing completion (Section B.2 explains their principal details).

The primary objectives of these tank tests is to ascertain how the device performs in respect of the following criteria :

1. Efficiency of energy capture;
2. Motions during working cycles;
3. Response to extreme waves;
4. Reactions to rode or power takeoff failure.

This information must be collected over a representative range of wave and device parameter values, as detailed below. It will show how the energy output relates to incident wave conditions, how the various components of the device then behave, and what the motions and forces are which it must bear in the design storm event. Taken together this information will be integrated with annual wave spectra to give both the fatigue and severe load and displacement spectra needed for the selection of components, hence the device may be optimised against the basic parameter values of the system. These are :

FIG.B.1.

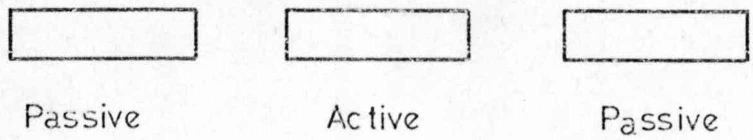


FIG.B.1a

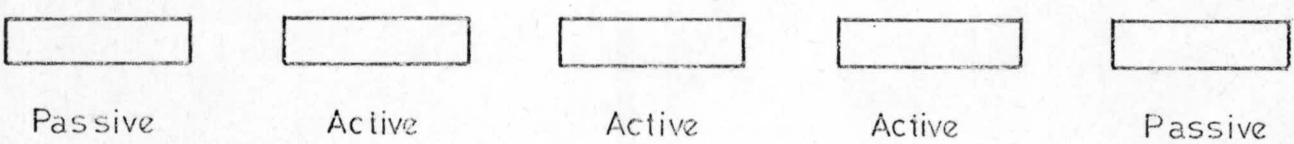


FIG.B.1b

1. Wave climate, including wave height, wavelength, direction with respect to cylinder alignment, short-crestedness and water depth. The bed profile around and seawards of the location in question may also influence the local wave climate.

2. Currents, both tidal and residual.

3. Cylinder diameter, length, end curvature, submergence, spacing, water depth and tuned period. Other parameters like the spring rate, unrestrained piston stroke, charactristics of the cushions, occurrence of slack in the rodes and the pumping pressure are also very important.

The first stage of testing (may-August 1980) will initially use one 'active' cylinder model to compare and extend the results obtained with one much simpler cylinder in the Edinburgh tank last year (see our Oct. '79 Report, Section 2). The following parameters must be varied.

1. Wave height, length and direction for regular waves for each of a range of;

2. Cylinder aspect ratios (length/diameter) from 2:1 to 6:1, but particularly 3:1 to 5:1, and;

3. Submergences from zero to 5m below mean sea level in waves, to identify the effect of submergence in general and tides in particular, and;

4. Water depth, from 40m to 70m, and;

5. Damping, to give optimum device performance.

An estimated total of 240 tests will be needed to identify the optimum values for these parameters over a representative range of wave conditions. This information will serve as a basis for the study of an array of three cylinders (Fig. B.1a), when the following parameter will also be considered :

6. Spacing between cylinders, from 5m to 50m, to show how lateral energy capture (i.e., device efficiency) changes (Section 2.5).

A total of about 200 tests will be needed to include this parameter and explore the validity for the three cylinder array of data collected using one cylinder in isolation.

It is expected that this stage of the laboratory work will give a good first understanding of each of the primary objectives of the tank tests listed above, and will be sufficient to enable other parts of the overall programme for Phase 4 to proceed.

The second stage of the laboratory work (late 1980-mid 1981) will establish cylinder performance in wave conditions representing those off S. Uist, including reference to mixed sea conditions and water depth. An array of five cylinders will then be used, the centre three of which will be of the 'active' type (Fig. B.1b). This will allow the performance of the central cylinder to be recorded when the adjacent cylinders are also under full control. The results will be compared with those determined using three cylinders, to show in more detail the importance of grouping.

At that stage any worthwhile changes to the design of the device arising from the wider optimisation study carried out up to that time will be incorporated into the models and the range of tests adjusted as necessary. This range will include :

1. Representative regular normal and oblique waves to compare the performances of the central cylinders of the three and five cylinder arrays, including making adjustments to identify optimum parameter values for the five cylinder array if these prove to be different from those for three cylinders;

2. Recording the capture efficiencies, mooring forces and cylinder orbits of the central cylinder in a representative selection of the 399 'Selected Set' S. Uist wave spectra, with each at an appropriate inclination to the optimum orientation of the cylinders. These 399 spectra are themselves only representative of the annual sequence, hence to choose less than 50 for the present purposes would be unwise.

3. Assessing how a limit applied to the cylinder orbit in high waves will affect the total capture efficiency of the device and the forces to be carried by the moorings and anchors. The pattern of pumping during a wave cycle in these conditions will be studied, together with the corresponding effects of slack rodes.

We expect that it will be necessary to carry out check tests to show whether the optimum values for aspect ratio, submergence, water depth, damping and spacing as determined in regular waves are appropriate in mixed seas. The following factors must also be considered at that stage ;

1. Peak mooring forces and side sway in steep waves (as a function of wavelength - see Section 10);

2. Rode angles at other than 45° and with inward and outward splay (Section 3.4 and Fig. 3.3).

Because of the qualifications made about currents in Section 10 and the results presented in Sections 2.3d and 2.3e about the possible (small) effect of these that, we calculate, will in general be experienced by the cylinder, we do not believe it is essential to study the performance of the cylinder in combined waves and currents as part of the programme for Phase 4. However, if engineering design studies follow in 1982, it will be necessary at that stage to consider whether certain aspects of the system may be sensitive to currents. This could involve further tank testing.

The second series of tests will include about 400 individual experiments, mostly with complex seas.

It is also intended that some preliminary studies should be made of cylinder behaviour in those failure modes that, by that time, appear most critical to the safety of the device and its neighbours (Section 2.7). These tests will be for general information at that stage; more comprehensive studies would be needed as part of engineering design work carried out in 1982.

B.2 The Models

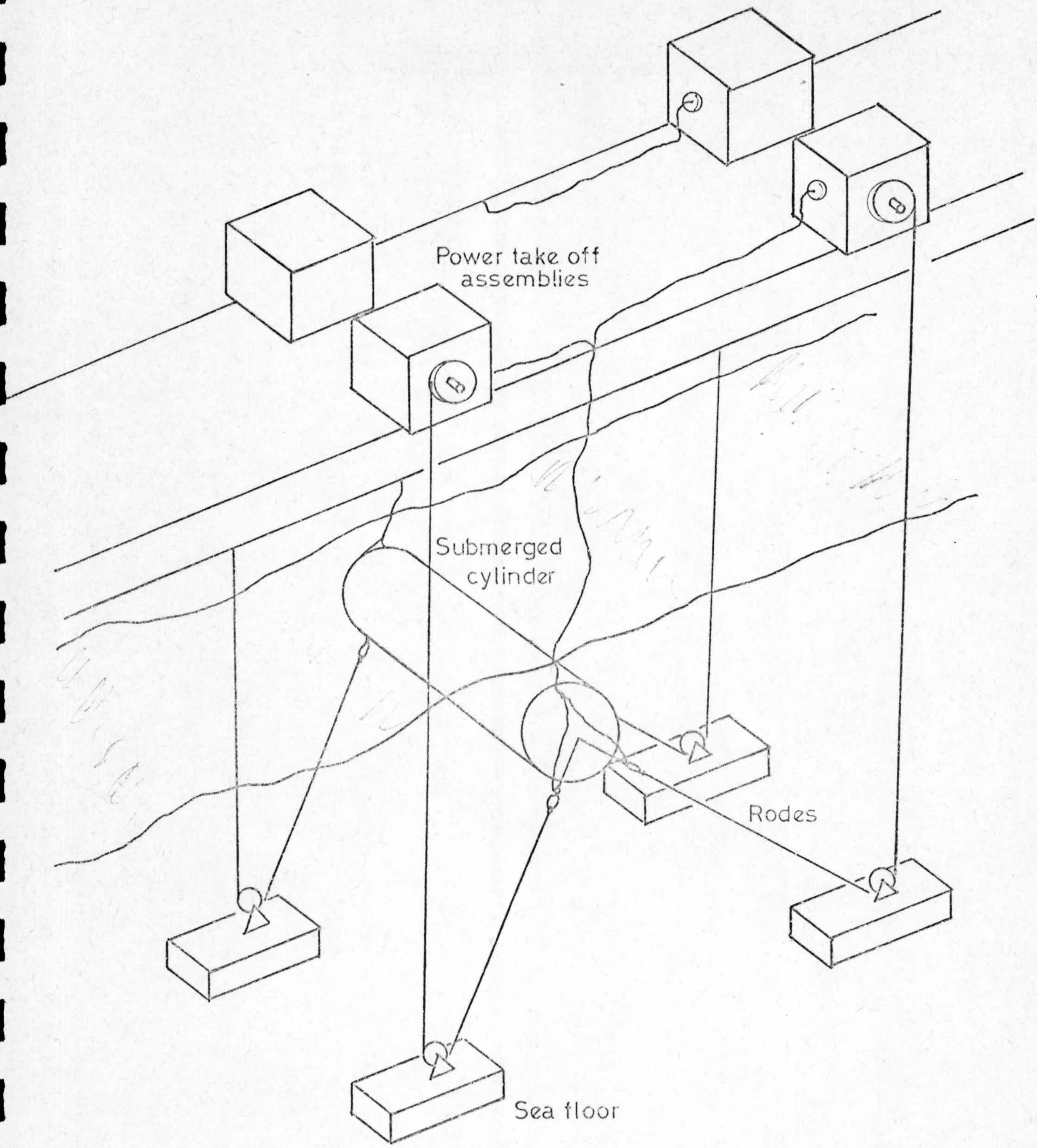
Each cylinder test unit is made up as shown in Fig. B.2. There are four corner power takeoff assemblies connected by 4 No. 32/9 Trace chains via 4 No. pulleys fixed at the appropriate water depth (Section 8.5) to the cylinder.

The corner power takeoff assembly is shown in Figs. B.3 and B.4, and this is connected into the control unit for the rig (Fig. B.5).

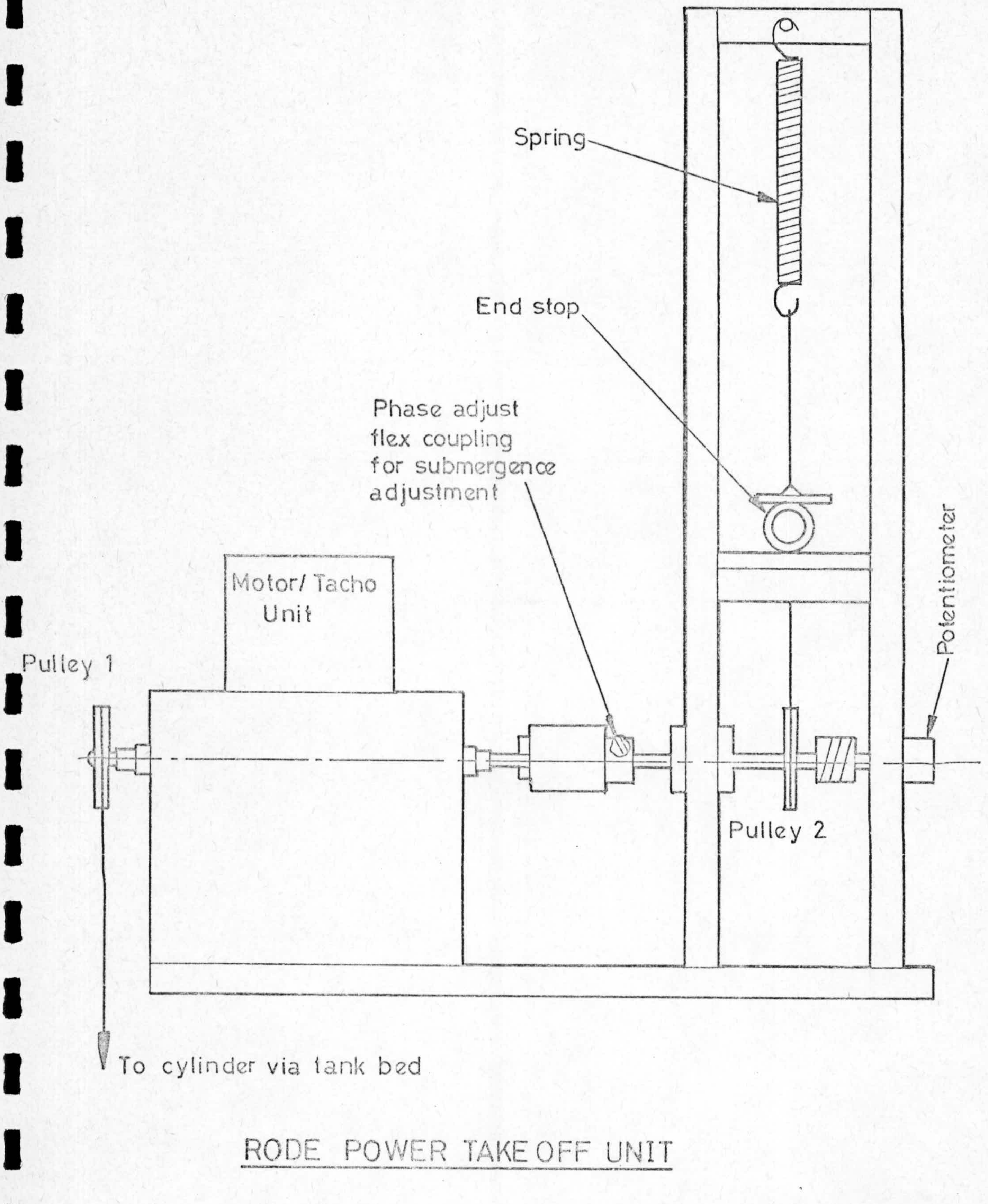
The 'active' cylinder model (two more will be made for the second test series) is fully instrumented and are controllable to give variable spring and damper characteristics. Rode forces, displacements, velocities and powers may be monitored continually, as detailed below.

The springs are basically mechanical with a fine electronic tuning. Two types are provided, namely linear and non-linear (as in our 'Edinburgh' tests, May 1979). Both types have adjustable end-stop facilities to simulate full-scale buffer effects.

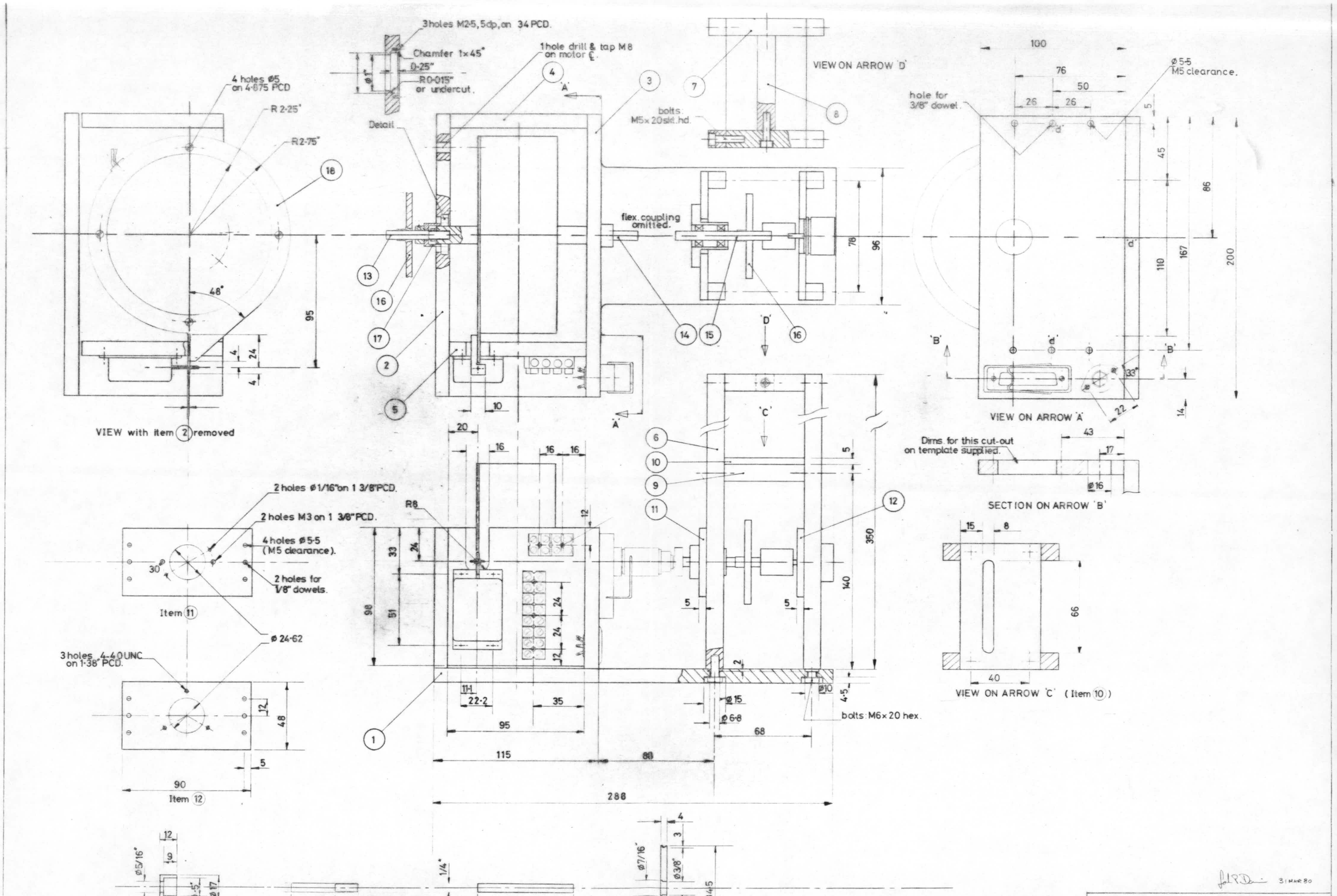
FIG.B.2.



CYLINDER TEST UNIT



RODE POWER TAKE OFF UNIT



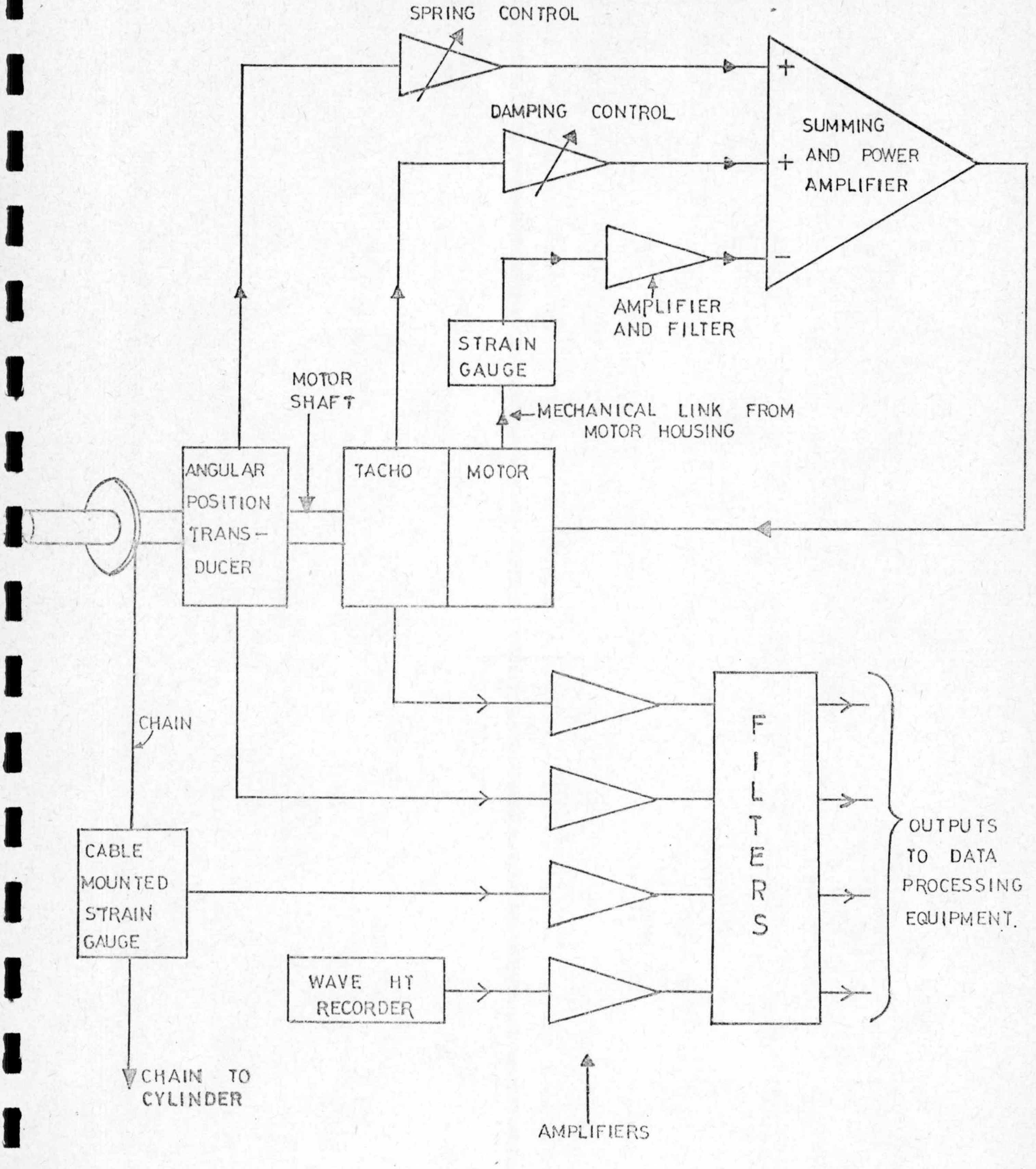
Submerged Cylinder

Wave Energy Project

Fig. B.4

CADNAM FRAME	01
All dimensions 'mm' except where stated	
CONSULT ITEMS LIST	DO NOT SCALE

FIG.B.5


SPRING CONTROL
DAMPING CONTROL
SUMMING AND POWER AMPLIFIER
+
+
-
AMPLIFIER AND FILTER
STRAIN GAUGE
MOTOR SHAFT
MECHANICAL LINK FROM MOTOR HOUSING
ANGULAR POSITION TRANS-DUCER
TACHO
MOTOR
CHAIN
FILTERS
CABLE MOUNTED STRAIN GAUGE
WAVE HT RECORDER
OUTPUTS TO DATA PROCESSING EQUIPMENT.
CHAIN TO CYLINDER
AMPLIFIERS

Damping is provided by motor units. For the first tests the damping will be of two types, namely linear ('viscous' or 'velocity proportional') and non-linear ('Coulomb' or 'bang-bang'), both of which may be varied continually. For the second part of the test programme the motor control unit will have been developed to include complex control philosophies using digital electronics.

The load/displacement characteristics of particular pump/spring combinations can be stored on Read Only Memories (ROM's). When these characteristics are read into the motor control unit the load applied to the rode at its power takeoff end is precisely regulated.

As the cylinder moves and transmits motion to the motor shaft via the chains and pulleys, so voltages are generated by the tachometer and angular position transducer. These voltages are amplified via the damping and spring rate controls respectively, and fed to the input of the summing amplifier; they may be regarded as 'demand' signals.

The summing amplifier drives a power amplifier, which in turn drives the motor such that it produces a force acting against the motion of the cylinder. The force produced by the motor is measured by a strain gauge mounted between the motor housing and the framework of the rig. The voltage derived from the strain gauge is amplified and filtered before being applied to the summing amplifier. Filtration is required to prevent system instability at the resonant frequency of the motor/strain gauge combination.

This 'closed loop' system, which toninually compares the output (in this case the force on the strain gauge) with the demanded inputs, and attempts to minimise the difference between demand and output, may thus be seen to be a typical servo system.

Fig. B.5 also shows the instrumentation built to acquire data from the model. Each transducer (angular position transducer, tachometer, cable mounted strain gauge) and wave height recorder produces a voltage which is amplified to a suitable level for supply to the data processing equipment, and filtered to reduce the noise level of the signal.

As the required signals are of such low frequency (10 Hz), the filters have been designed to attenuate such frequencies as 50 Hz and above, while introducing small phase shifts at the signal frequencies. The amplified and filtered voltages are fed to the data processing equipment for analysis of the cylinder's performance.

The power supply provides 4 No. 30 volt unregulated supplies to drive each of the motors, and two regulated supplies, one of 5v and one of 15v, for driving the motor control electronics and the instrumentation.

It was found that commercially available strain gauge amplifiers drifted unacceptably with temperature. The units that have been made up for this purpose incorporate all the worthwhile features of the commercial ones together with refinements to improve performance.

Buffer amplifiers with switched variable gain are provided so that suitably large signals may be fed to the data processing equipment. The angular position transducer amplifiers provide a 'level shift' as well as amplification so that voltages due to the static position of the motor/A.P.T. may be removed; this allows only the signal caused by movement of the A.P.T. to be amplified.